



Finanziato  
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NextGenerationEU



Ministero  
dell'Università  
e della Ricerca



Italiadomani  
PIANO NAZIONALE  
DI RIPRESA E RESILIENZA



UNIVERSITÀ  
DEGLI STUDI  
FIRENZE

# DOTTORATO DI RICERCA IN

## *Ingegneria Industriale*

CICLO XXXVIII - ex D.M. 352/2022

### *Small-Scale Compressed Air Energy Storage: a Techno-Economic and Environmental Perspective*

Settore Scientifico Disciplinare

ING-IND / 09

#### **Dottorando**

Dott. Tumminello Dario

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#### **Supervisore**

Prof. Facchini Bruno

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#### **Coordinatore**

Prof. Governi Lapo

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Via di Santa Marta, 3, 50139 Firenze, Italy.

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# Acknowledgements

*Giunti a questo punto del viaggio, sebbene questo dottorato sia stato un viaggio piuttosto solitario, è doveroso fare alcuni ringraziamenti.*

*Innanzitutto un grazie al Prof. Facchini, per avermi ricercato dopo che avevo lasciato l'università, per avermi dato la possibilità di fare questa esperienza e per l'attenzione dimostrata verso tutti noi dottorandi in questi anni.*

*Un sincero ringraziamento va all'Ing. Tommaso Bacci (o, più semplicemente, Tommi) per il costante supporto e la pazienza con cui mi ha accompagnato in questi anni lungo un percorso ben lontano dal suo principale ambito di competenza, e per avermi trasmesso il valore del rigore scientifico nell'affrontare questo lavoro.*

*Un ringraziamento a tutti i ragazzi dell'HTC-group; un primo grazie speciale a Nicco e Umbe con cui ho condiviso il magico mondo del laboratorio Colonne e con cui è nato un sincero rapporto di amicizia. Grazie a tutti gli altri: Isa, Gian, Ale ec.. (tutti siamo troppi) per aver reso piacevole l'ambiente lavorativo e perché in questi anni siamo stati un gruppo di amici e non solo dei colleghi...anche se come ben sappiamo io sono sempre stato palesemente un infiltrato in questo gruppo di ricerca.*

*Grazie a Claudio per l'estrema disponibilità e simpatia dimostrata nell'aiutarmi con le analisi ambientali contenute nella tesi e perché, nonostante non fossimo dello stesso gruppo di ricerca, è stato probabilmente la persona con cui ho collaborato di più in questo ultimo periodo.*

*Un ringraziamento anche a chi ha reso questo dottorato un po' più internazionale, al Prof. Sciacovelli, per avermi permesso di svolgere*

*un periodo di ricerca presso l'University of Birmingham e per avermi insegnato a strutturare una ricerca nel modo più coerente possibile. Thanks to all the "survivors": Lorenzo, Hassan, Mostafa, Raman, Tomek and the others. You have been incredibly welcoming from the very first day, and even though we did not collaborate from a working point of view, it was a real pleasure to share the office with you. Thanks also to my Turkish mates, Bader and Utku, for being much more than just flatmates during those three months. I truly hope to see you all again.*

*I would also like to thank the solar-thermal group at CENER in Pamplona: first of all, Fritz, for hosting me within his research group. A special thanks goes to Javi, probably one of the very few people in the world able to fully understand all my pain with CAES and Modelica, it has been a real pleasure to work with you and to share enjoyable moments together. Thanks, as well to all the other colleagues in Pamplona: Ruben, Xabier, Asier, and all the others for welcoming me into their group as if I had always been part of it.*

*Un ringraziamento poi a tutti quelli che non sono entrati direttamente in questo viaggio di dottorato ma hanno contribuito con loro supporto dall'esterno.*

*Grazie alla mia famiglia, per l'incondizionato supporto che mi ha sempre dato da trent'anni a questa parte e che mi ha accompagnato anche quando ho lasciato un lavoro per iniziare questa pazzia chiamata dottorato. A mia mamma, ai miei fratelli a mia nonna.*

*Grazie ai miei amici, in primis meri (sì scritto così) e Simo, per aver condiviso momenti divertenti, ma anche momenti di fatica, che sono quelli che faccio sempre più fatica a condividere. Grazie a tutti gli altri, scout, ciclisti e ogni altra categoria di amicizie che mi hanno accompagnato in questi anni da ben prima di pensare di fare un dottorato.*

*Grazie infine ad Anna, per essermi stata vicina in quest'ultimo periodo senza essersi fatta spaventare dal mio imbrutimento da dottorato (in realtà già da quando ero nella triste e piovosa Inghilterra), e per riempire le mie giornate di risate e complicità.*

# Publications

## International Journals

- Tumminello, D., Zuffi, C., Bacci, T. and Facchini, B. "Hybrid micro-compressed air energy storage with photovoltaic and diesel generation for off-grid applications: A parametric life cycle and techno-economic study". *Cleaner Energy Systems*, 100246 (2026). <https://doi.org/10.1016/j.cles.2026.100246>
- **Manuscript under review:** Tumminello, D., Bacci, T., Facchini, B., and Sciacovelli, A. "Impact of Load Variability on Micro-CAES Systems: Assessment of Component and System Performance" Submitted to "*Energy Conversion and Management*".

## International Conferences

- Tumminello D, Bacci T. and Facchini B. "Numerical modeling and exergetic analysis of efficiency improving solutions for a micro-CAES system." *Journal of Physics: Conference Series*. Vol. 2648. No. 1. IOP Publishing, 2023. DOI 10.1088/1742-6596/2648/1/012045
- Tumminello D, Bacci T. and Facchini B. "Numerical Modeling and Design Optimization of a Micro-CAES System" 37th International Conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy Systems, ECOS 2024. Vol. 1. ECOS 2024, 2024. <https://doi.org/10.52202/077185-0003>

- Tumminello D, Bacci T., Facchini B. and Sciacovelli A. "Impact of Load Variability on Micro-CAES Systems: Assessment of Component and System Performance" 38th International Conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy Systems, ECOS 2025
- Tumminello D, Bacci T., Baigorri J. and Facchini B. "Integration of Radial Turbomachinery in Small-Scale Compressed Air Energy Storage: Impact of System Size on Performance and Cost-Effectiveness" accepted for 39th International Conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy Systems, ECOS 2026

# Abstract

The increasing penetration of variable renewable energy sources required to meet international decarbonization targets, such as those established by the Paris Agreement, calls for energy storage technologies capable of providing flexibility, reliability, and long-duration energy shifting. While electrochemical batteries are widely deployed for short-term storage and fast-response applications, they cannot represent the sole solution to global energy storage needs. In particular, the large-scale deployment of stationary battery systems may be constrained by the limited availability and production capacity of critical raw materials. Consequently, the development of a diversified portfolio of energy storage technologies is essential. Within this context, Compressed Air Energy Storage (CAES) represents a potentially attractive option, especially for decentralized and resource-constrained energy systems.

This thesis presents a comprehensive assessment of small-scale CAES systems, focusing on realistic system architectures based on commercially available components and operated under time-varying conditions. A dynamic modeling framework was developed in the OpenModelica environment and validated against experimental data obtained from an industrial micro-CAES prototype.

Building on this framework, the analysis was extended through an integrated sensitivity analysis and surrogate-based optimization approach implemented using the Dakota toolkit. This methodology enabled a systematic exploration of the design space and an evaluation of the influence of key design parameters on overall system performance. The results

show that small-scale CAES systems are generally characterized by moderate round-trip efficiencies, typically in the range of 30–40%. Moreover, the analysis demonstrates that realistic load variability has a significant impact on system efficiency, with reductions of up to 10–15 percentage points observed under highly fluctuating demand profiles. These findings highlight the importance of accounting for realistic operating conditions in the design and assessment of CAES systems.

To assess the feasibility of small-scale CAES, this thesis presents a comprehensive techno-economic and life-cycle assessment of micro-CAES systems based on standard volumetric machines, integrated into hybrid photovoltaic–diesel microgrids for remote and off-grid communities. Parametric life cycle inventories were developed as a function of system size, enabling consistent and scalable environmental evaluations across different configurations and contributing to fill the gap in environmental assessments for small-scale CAES identified in the literature. The results indicate that the integration of CAES can substantially reduce greenhouse gas emissions, with CO<sub>2</sub> emissions decreasing by more than 60% compared to systems where photovoltaic generation is not coupled with energy storage. However, economic competitiveness remains strongly dependent on local solar resource availability and fuel prices. In the locations analyzed, the levelized cost of energy associated with storage integration ranges from an increase of approximately 0.03 \$/kWh to a reduction of about 0.05 \$/kWh, with the most favorable outcomes observed in scenarios characterized by high solar availability and elevated fuel costs.

Alternative system configurations based on radial turbomachinery were also investigated to assess their potential at small scale. The analysis indicates that turbomachinery-based solutions may achieve higher round-trip efficiencies, on the order of 50–60%, and lower storage costs when deployed at sufficiently large system sizes. However, a scale-dependent transition is identified, below which performance degradation and economic penalties limit their applicability.

The analysis is finally extended to grid-connected small-scale CAES systems to assess their role under variable electricity market conditions.

Price-arbitrage-oriented operating strategies are evaluated as a function of system size and market dynamics. The results show that price arbitrage alone is generally insufficient to ensure economic viability for small- and medium-scale CAES systems, which becomes achievable only in electricity markets with extreme intraday price spreads or when combined with cost-reduction measures and regulatory incentives. These findings point to the need for alternative operational paradigms to improve system profitability.

Overall, this work provides an integrated modeling, optimization, and life-cycle assessment framework for the analysis of small-scale CAES systems, offering quantitative insights into their performance, environmental impacts, and design trade-offs. By explicitly accounting for dynamic operation, off-design component behavior, and realistic operating conditions, the thesis contributes to a clearer understanding of the potential role of micro-CAES in decentralized energy systems with high shares of variable renewable energy sources.





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# Nomenclature

*Note: some symbols are used with different meanings depending on the context*

## Acronyms

|         |                                                  |                |
|---------|--------------------------------------------------|----------------|
| AA-CAES | Advanced Adiabatic Compressed Air Energy Storage |                |
| A-CAES  | Adiabatic Compressed Air Energy Storage          |                |
| BES     | Battery Energy Storage                           |                |
| CAES    | Compressed Air Energy Storage                    |                |
| CAPEX   | Capital Expenditure                              |                |
| CAS     | Compressed Air Storage                           |                |
| CEPCI   | Chemical Engineering Plant Cost Index            |                |
| CES     | Chemical Energy Storage                          |                |
| CF      | Cash flow                                        | [\$]           |
| CHP     | Combined Heat and Power                          |                |
| CSP     | Concentrating Solar Power                        |                |
| D-CAES  | Diabatic Compressed Air Energy Storage           |                |
| DG      | Diesel Generator                                 |                |
| EF      | Environmental Footprint method                   |                |
| EGO     | Efficient Global Optimization                    |                |
| ER      | Energy resources depletion                       | MJ/kWh         |
| FBES    | Flow Battery Energy Storage                      |                |
| FE      | Freshwater eutrophication                        | [kg P eq./kWh] |
| FES     | Flywheel Energy Storage                          |                |
| FET     | Freshwater eco-toxicity                          | [CTUe/kWh]     |



|        |                                             |                               |
|--------|---------------------------------------------|-------------------------------|
| FIP    | Feed-In Premium                             |                               |
| FMI    | Functional Mock-up Interface                |                               |
| GA     | Genetic Algorithm                           |                               |
| GWP    | Global warming potential                    | [kg CO <sub>2</sub> eq. /kWh] |
| HEX    | Heat Exchanger                              |                               |
| HP     | Horse power                                 |                               |
| HTC    | Human toxicity carcinogenic                 | [CTUh/kWh]                    |
| HTNC   | Human toxicity non-carcinogenic             | [CTUh/kWh]                    |
| I-CAES | Isothermal Compressed Air Energy Storage    |                               |
| IGV    | Inlet guide vane                            |                               |
| IR     | Ionization radiation                        | [kBq U-<br>235eq./kWh]        |
| LAES   | Liquid Air Energy Storage                   |                               |
| LCA    | Life Cycle Assessment                       |                               |
| LCI    | Life Cycle Inventory                        |                               |
| LCOE   | Levelized cost of energy                    | [\$/kWh]                      |
| LCOS   | Levelized Cost of Storage                   | [\$/kWh]                      |
| LDES   | Long-Duration Energy Storage                |                               |
| LHS    | Latent Heat Storage                         |                               |
| LHV    | Lower heating value                         | [J/kg]                        |
| LU     | Land use                                    | [-]                           |
| ME     | Marine eutrophication                       | [kg N eq./kWh]                |
| MOGA   | Multi objective Genetic Algorithm           |                               |
| MR     | Mineral resource depletion                  | [kg Sb eq/kWh]                |
| NPV    | Net present value                           | [\$]                          |
| NTU    | Number of transfer units                    | [-]                           |
| O&M    | Operation and Maintenance cost              | [\$]                          |
| OD     | Ozone depletion                             | [kg CFC-11eq. /kWh]           |
| ORC    | Organic Rankine Cycle                       |                               |
| PCMs   | Phase Change Materials                      |                               |
| PHES   | Pumped Hydroelectric Energy Storage         |                               |
| PI     | Proportional-Integral controller            |                               |
| PID    | Proportional-Integral-Derivative controller |                               |

|         |                                           |                          |
|---------|-------------------------------------------|--------------------------|
| PM      | Particulate matter formation              | [disease incidence /kWh] |
| POF     | Photochemical oxidant formation           | [kg NMVOC eq. /kWh]      |
| PR      | Performance Ratio                         |                          |
| PTES    | Pumped Thermal Energy Storage             |                          |
| PV      | Photovoltaic                              |                          |
| RMSE    | Root mean square error                    | [-]                      |
| RSS     | Renewable Self-Sufficiency                | [%]                      |
| RTE     | Round-Trip Efficiency                     | [%]                      |
| SBO     | Surrogate based optimization              |                          |
| SCES    | Supercapacitors Energy Storage            |                          |
| SHS     | Sensible Heat Storage                     |                          |
| SNG     | Synthetic Natural Gas                     |                          |
| SOC     | State of charge of storage system         | [-]                      |
| SS      | Single Score environmental indicator      | [pt/kWh]                 |
| SS-CAES | Small-scale compressed air energy storage |                          |
| TCES    | Thermochemical Energy Storage             |                          |
| TE      | Terrestrial eutrophication                | [mol N eq./kWh]          |
| TES     | Thermal Energy Storage                    |                          |
| USD     | US-dollar                                 |                          |
| VSD     | Variable Speed Drive                      |                          |
| VSV     | Variable stator vane                      |                          |
| WU      | Water use                                 | [m <sup>3</sup> ]        |

### Greeks

|                   |                                                       |                      |
|-------------------|-------------------------------------------------------|----------------------|
| $\alpha$          | Angle (turbomachinery)                                | [°]                  |
| $\alpha$          | Convective heat transfer coefficient (thermodynamics) | [W/m <sup>2</sup> K] |
| $\beta$           | Pressure ratio                                        | [-]                  |
| $\bar{\beta}$     | Normalized pressure ratio                             | [-]                  |
| $\delta$          | Wall thickness                                        | [m]                  |
| $\varepsilon$     | Heat exchanger effectiveness                          | [-]                  |
| $\eta$            | Generic efficiency                                    | [-]                  |
| $\bar{\eta}_{is}$ | Normalized isentropic efficiency                      | [-]                  |

|                |                                 |                      |
|----------------|---------------------------------|----------------------|
| $\eta_{is}$    | Isentropic efficiency           | [-]                  |
| $\eta_{vol}$   | Volumetric efficiency           | [-]                  |
| $\gamma$       | Heat capacity ratio             | [-]                  |
| $\gamma_A$     | Absolute waste exergy ratio     | [-]                  |
| $\gamma_L$     | Local waste exergy ratio        | [-]                  |
| $\nu$          | Kinematic viscosity             | [m <sup>2</sup> /s]  |
| $\nu^*$        | Modified mean                   | [-]                  |
| $\omega$       | Angular velocity                | [rad/s]              |
| $\omega_s$     | Specific speed                  | [-]                  |
| $\Phi$         | Flow coefficient                | [-]                  |
| $\rho$         | Density                         | [kg/m <sup>3</sup> ] |
| $\sigma$       | Variance (statistics)           | [-]                  |
| $\sigma$       | Stress (mechanical engineering) | [MPa]                |
| $\sigma_{all}$ | Allowable stress                | [MPa]                |
| $\sigma_y$     | Yield stress                    | [MPa]                |
| $\Psi$         | Load coefficient                | [-]                  |

### Letters

|          |                                             |                   |
|----------|---------------------------------------------|-------------------|
| $A$      | Area                                        | [m <sup>2</sup> ] |
| $A_c$    | Annual cost                                 | [\$]              |
| $A_r$    | Aspect ratio                                | [-]               |
| $AT$     | Acidification                               | [mol H+ eq/kWh]   |
| $C$      | Cost (economic)                             | [\$]              |
| $C$      | Heat capacity rate (thermodynamics)         | [W/K]             |
| $C_r$    | Thermal capacity ratio                      | [-]               |
| $c_{el}$ | Electricity unit cost                       | [\$/kWh]          |
| $c_p$    | Specific heat capacity at constant pressure | [J/kgK]           |
| $c_v$    | Specific heat capacity at constant volume   | [J/kgK]           |
| $d$      | Discount rate                               | [-]               |
| $D$      | Diameter                                    | [m]               |
| $D_s$    | Specific diameter                           | [-]               |
| $E$      | Energy                                      | [J]               |
| $Ex$     | Exergy                                      | [J]               |

|             |                                       |                             |
|-------------|---------------------------------------|-----------------------------|
| $ex$        | Specific exergy                       | [J/kg]                      |
| $F_p$       | Pressure correction factor            | [-]                         |
| $h$         | Specific enthalpy                     | [J/kg]                      |
| $I$         | Irradiance                            | [W/m <sup>2</sup> ]         |
| $I_{STC}$   | Irradiance at standard test condition | [1000 W/m <sup>2</sup> ]    |
| $k$         | Thermal conductivity                  | [W/mK]                      |
| $l$         | Length                                | [m]                         |
| $m$         | Mass                                  | [kg]                        |
| $\dot{m}$   | Mass flow rate                        | [kg/s]                      |
| $M_r$       | Reduced mass flow rate                | [kg K <sup>1/2</sup> /s Pa] |
| $\bar{M}_r$ | Normalized reduced mass flow rate     | [-]                         |
| $N$         | Rotational speed                      | [rpm]                       |
| $N_r$       | Reduced rotational speed              | [rpm/ K <sup>1/2</sup> ]    |
| $\bar{N}_r$ | Normalized reduced rotational speed   | [-]                         |
| $p$         | Pressure                              | [Pa]                        |
| $P$         | Power                                 | [W]                         |
| $Q$         | Heat                                  | [J]                         |
| $\dot{Q}$   | Heat transfer rate                    | [W]                         |
| $R$         | Gas constant (thermodynamics)         | [J/kgK]                     |
| $R$         | Revenue (economic)                    | [\$]                        |
| $Re$        | Reynolds number                       | [-]                         |
| $s$         | Specific entropy                      | [J/kgK]                     |
| $t$         | Time                                  | [s]                         |
| $T$         | Temperature                           | [K]                         |
| $u$         | Specific internal energy              | [J/kg]                      |
| $U$         | Overall heat transfer coefficient     | [W/m <sup>2</sup> K]        |
| $UA$        | Overall heat transfer conductance     | [W/K]                       |
| $v$         | Specific volume                       | [m <sup>3</sup> /kg]        |
| $V$         | Volume                                | [m <sup>3</sup> ]           |
| $V_s$       | Swept volume                          | [m <sup>3</sup> /rev]       |
| $w$         | Specific work                         | [J/kg]                      |
| $W$         | Work                                  | [J]                         |
| $Y$         | System lifetime                       | [year]                      |

**Sub and Superscripts**

|             |                                   |
|-------------|-----------------------------------|
| 0           | Thermodynamic reference condition |
| 1           | Inlet condition                   |
| 2           | Outlet condition                  |
| <i>ad</i>   | Adiabatic                         |
| <i>amb</i>  | Ambient conditions                |
| <i>c</i>    | Compressor                        |
| <i>C</i>    | Cold side                         |
| <i>ch</i>   | Charging                          |
| <i>comp</i> | Compressor                        |
| <i>D</i>    | Destruction                       |
| <i>dis</i>  | Discharging                       |
| <i>el</i>   | Electrical                        |
| <i>ex</i>   | Expander                          |
| <i>H</i>    | Hot side                          |
| <i>in</i>   | Inlet                             |
| <i>L</i>    | Loss                              |
| <i>L</i>    | Load                              |
| <i>max</i>  | Maximum admissible value          |
| <i>min</i>  | Minimum admissible value          |
| <i>n</i>    | Nominal (design-point) condition  |
| <i>opt</i>  | Optimal condition                 |
| <i>out</i>  | Outlet                            |
| <i>PV</i>   | Photovoltaic                      |
| <i>ref</i>  | Reference value                   |
| <i>st</i>   | Storage                           |
| <i>TA</i>   | Thermodynamic average             |

# Chapter 1

## Introduction

### 1.1 Context

Global energy systems are undergoing a profound transformation driven by the urgent need to mitigate anthropogenic climate change and to comply with international commitments targeting carbon neutrality. In this context, 195 nations agreed under the Paris Agreement to limit the global average temperature increase to well below 2 °C and to achieve net global greenhouse gas (GHG) emission neutrality by 2050 [1]. Achieving these targets requires deep decarbonization of the energy system, with particularly significant reductions in emissions from the power generation sector worldwide [2].

The large-scale deployment of renewable energy sources represents a cornerstone of this transition, enabling a progressive reduction in GHG emissions associated with electricity generation [3, 4]. According to the International Energy Agency (IEA), global renewable energy capacity is expected to increase by nearly 4,600 GW between 2025 and 2030, corresponding to a growth rate twice as high as that observed over the 2019–2024 period. As a result, renewable electricity generation could rise by approximately 60% between 2024 and 2030, increasing from about 9,900 TWh to more than 16,000 TWh [5].

However, the intrinsic intermittency and limited predictability of re-

newable sources such as wind and solar introduce significant challenges for power system reliability, stability, and operational flexibility. As renewable penetration increases, mismatches between electricity generation and demand become more frequent, particularly in systems characterized by high shares of non-dispatchable resources. Addressing these mismatches requires the adoption of energy storage systems capable of temporally decoupling electricity production from consumption, thereby ensuring continuity of supply and supporting grid stability [6, 7]. In this sense, energy storage is widely recognized as a key enabling technology for the decarbonization of the energy sector and has been explicitly identified by the European Commission as a strategic pillar for achieving long-term climate objectives [4]. Recent IEA assessments further highlight this trend, estimating that increasing global renewable energy capacity by 2030 will require approximately 1,500 GW of energy storage capacity worldwide [8].

Beyond large, interconnected transmission networks, the role of energy storage becomes even more critical in small-scale and decentralized applications. Microgrids, stand-alone systems, and remote or rural communities increasingly rely on locally available renewable generation, such as photovoltaic panels or small wind turbines, to meet their electricity demand. In these contexts, storage systems are essential not only to enhance operational flexibility but also to guarantee energy access, supply security, and resilience against generation variability [9]. In many remote and islanded areas, electricity supply still relies heavily on diesel generators, which are commonly used either as the primary energy source or as backup units to compensate for renewable intermittency. For instance, Weis and Ilinca [10] report that in Canada approximately 200,000 people depend on diesel-generated electricity, resulting in annual emissions of about 1.2 million tons of greenhouse gases. Reducing dependence on diesel generation therefore represents a critical objective for mitigating the environmental impact of energy systems in remote communities. In this regard, the combination of renewable energy sources with adequate energy storage constitutes a key pathway to displace fossil fuel-based

generation while ensuring reliable electricity supply [11].

## 1.2 Motivation

### 1.2.1 The need for diversified energy storage solutions

In this context, a wide range of electrical energy storage (EES) technologies are currently under development or deployment, spanning different power and energy scales and characterized by varying levels of technological maturity [12]. Electrochemical batteries, particularly lithium-ion systems, currently dominate the small and medium-scale storage market due to their high efficiency coupled with high energy density and fast response. Nevertheless, their large-scale deployment as the primary storage solution poses several challenges.

Lithium-ion battery technologies rely on raw materials that are considered critical, including lithium, cobalt, nickel, and graphite [13]. The extraction and processing of these materials are geographically concentrated and often associated with environmental degradation, social concerns, and geopolitical risks. Lithium resources, for instance, are unevenly distributed, with a significant fraction located in South America, while cobalt mining has been widely linked to severe environmental pollution, unsafe working conditions, and ethical issues such as child labor [14, 15]. These challenges are further exacerbated by the slow deployment of large-scale recycling infrastructures and by strong competition for critical materials from other sectors, most notably electric mobility.

The limitations of battery-centric storage pathways become particularly evident when considering long-term decarbonization scenarios. Several resource-based assessments indicate that, even under optimistic assumptions regarding technological progress, recycling rates, and the exclusion of competing sectors, large-scale deployment of stationary battery storage may not be feasible due to insufficient reserves and production capacities of critical raw materials, particularly lithium and cobalt [14, 16, 17]. Analyses based on global energy transition scenarios suggest that the cumulative material demand associated with lithium-ion batteries



required to support high shares of renewable energy could exceed known reserves by mid-century, with constraints that cannot be fully alleviated by recycling alone. In addition, efforts to reduce the cobalt content of battery cells, often result in increased reliance on other potentially critical metals, such as nickel, thereby shifting rather than eliminating resource bottlenecks [14].

These findings suggest that, although batteries will continue to play a crucial role in future energy systems, especially for short-duration storage applications, they cannot be regarded as the sole solution for meeting global energy storage needs. Consequently, the development of a diversified portfolio of energy storage technologies based on different physical principles and material requirements appears necessary to ensure the scalability, sustainability, and robustness of low-carbon energy systems.

### 1.2.2 Small-scale compressed air energy storage potential

Within the broader framework of energy storage diversification, compressed air energy storage has re-emerged as a promising option within the portfolio of energy storage technologies [18, 19], particularly for applications requiring medium to long-duration storage. CAES systems store electrical energy in the form of pressurized air using electrically driven compressors and release it through expansion machines to regenerate electricity when needed [20]. Unlike electrochemical storage technologies, CAES relies primarily on mechanical components and does not require large quantities of critical raw materials, potentially mitigating material supply risks and environmental concerns.

Historically, CAES has been developed primarily for large-scale applications [21, 22], while more recently, increasing attention has been devoted to the development of small-scale CAES systems [23, 24]. Small-scale CAES systems, operating in the kW to MW range, are of particular interest for decentralized and stand-alone applications. Their mechanical nature enables long service life, high cycle durability, and limited performance degradation over time. Moreover, small-scale CAES systems are not constrained by specific geographical requirements, such as those

affecting pumped hydro storage, and can be integrated into a wide range of distributed energy systems. These characteristics make small-scale CAES an attractive option for microgrids, remote communities, and off-grid applications, where long-term reliability, robustness, and sustainability are often as important as round-trip efficiency [25].

### 1.2.3 Small-scale CAES limitations and literature gaps

Despite these advantages, small-scale CAES systems face several intrinsic limitations that currently hinder widespread deployment. Their round-trip efficiency is generally limited, particularly for diabatic or quasi-adiabatic layouts, with typical values in the range of 30–40% [26]. In addition, the low energy density of compressed air leads to relatively large system volumes and footprints, which may limit applicability in space-constrained environments. System performance is also strongly affected by thermal phenomena during compression and expansion, making thermal management a critical design challenge.

To overcome these limitations, a wide range of integration strategies has been proposed in the literature, spanning both small and large-scale applications [27]. These include the coupling of CAES with high-temperature thermal energy storage to replace conventional combustion chambers in diabatic configurations, the integration of Organic Rankine Cycles (ORC) for waste heat recovery, and hybridization with trigeneration systems, solar thermal collectors, concentrated solar power plants, or even desalination units [28–33]. While such integrations can significantly enhance overall efficiency and reduce fuel consumption or exergy destruction, they also increase system complexity and often require auxiliary resources or services (e.g., waste heat availability, water demand) that are not always present or justified in decentralized applications.

Alternative research efforts focus on approaching isothermal compression and expansion processes, which theoretically minimize compression work and thermal losses [34, 35]. These concepts are typically realized through prototype solutions such as water spray injection or liquid-piston machines [36–38]. However, such technologies remain largely at the ex-

perimental or prototype level and are not yet suitable for widespread commercial deployment.

Focusing on more conventional adiabatic or near-adiabatic CAES configurations, it is worth noting that, despite decades of research on CAES, the vast majority of studies address large-scale, utility-level installations. As highlighted by several recent reviews, the literature on small-scale CAES systems remains scarce, fragmented, and often incomplete [27]. In particular, comprehensive studies jointly addressing technology development, system integration, design methodologies, operational strategies, and optimization at the kW to MW scale are still lacking.

This gap is particularly critical given the typical operating conditions of small-scale CAES systems. Unlike large grid-connected plants, small-scale systems are often embedded in microgrids or stand-alone energy systems characterized by highly variable renewable generation and fluctuating loads. Moreover, many small-scale configurations rely on isochoric air storage, leading to continuously varying storage pressure during charging and discharging. As a result, these systems operate predominantly under off-design conditions rather than at nominal design points. Although recent contributions have begun to introduce more detailed component-based and dynamic modeling approaches, many existing studies still rely on steady-state or design-point analyses [27, 39]. Partial-load behavior is often investigated by evaluating performance at discrete fractions of nominal power [40–42], an approach that is of limited relevance for stand-alone applications, where system operation must continuously adapt to coupled load and generation profiles. For example, Guo et al. [43] analyzed exergy losses during start-up transients, while early micro-scale studies were largely based on idealized assumptions and simplified component model

Numerous recent contributions have explored the techno-economic potential of small-scale CAES, showing that micro-CAES systems may become competitive with battery-based storage for discharge durations exceeding approximately six hours [44, 45]. However, in most cases, these analyses still assume rated or quasi-steady operating conditions, and off-

design and transient behavior are not explicitly accounted for in long-term performance assessment. Similarly, sizing methodologies for stand-alone micro-CAES systems have been proposed [46], but the impact of load variability on long-term performance is rarely investigated in detail.

System sizing and optimization also remain major open challenges. The strong coupling between compressors, expanders, thermal storage, and air reservoirs makes appropriate component sizing essential to avoid efficiency losses and economic penalties [47]. Although several sizing and optimization approaches have been proposed, ranging from analytical formulations [48] to evolutionary algorithms [28, 49], only a limited number of studies attempt a fully integrated optimization of all major components, particularly under off-design operating conditions. Finally, the environmental performance of small-scale CAES systems is still insufficiently explored. While life cycle assessment (LCA) studies on energy storage technologies are increasing, environmental analyses of CAES remain less common and are mostly focused on large-scale installations [50–52]. Existing results show a wide variability in environmental impacts, strongly influenced by system efficiency and electricity mix. Recent findings also suggest that the integration of energy storage does not automatically reduce environmental impacts compared to renewable-only solutions, highlighting the need for application-specific and system-level assessments [53].

In light of the current state of the art, further research is clearly required to realistically assess the technical performance, economic viability, and environmental impact of small-scale CAES systems. In particular, there is a strong need for studies that move beyond idealized assumptions, explicitly account for off-design operation, and adopt integrated modeling and optimization frameworks capable of capturing the complex interactions between components, operating strategies, and application-specific constraints. Addressing these challenges constitutes the primary motivation for the present doctoral research.

### 1.3 Objectives and contribution of the thesis

In light of the considerations discussed above, this thesis aims to provide a comprehensive and realistic assessment of small-scale CAES systems, focusing on configurations that are technically mature and potentially deployable in the short term. While a significant portion of the existing literature investigates innovative or highly integrated CAES concepts (i.e. isothermal machines, hybrid systems with Organic Rankine Cycles, or multi-generation layouts) this work deliberately focuses on small-scale CAES architectures based on conventional and commercially available components. The objective is not to propose novel or idealized system configurations, but rather to understand what level of performance, efficiency, and sustainability can realistically be expected from technologically mature CAES systems when operated under real-world conditions.

To achieve this overarching goal, this doctoral research pursues a set of interconnected objectives that define both the scope of the work and its original scientific contributions. Each objective addresses one or more of the gaps identified in the literature on small-scale CAES systems and is associated with specific methodological or application-oriented advances developed within the thesis.

- **Objective:** To develop robust and systematic sizing and optimization methodologies for small-scale CAES systems and to assess their robustness and reliability under realistic operating conditions, explicitly accounting for strong component coupling and off-design operation.

**Contribution:** The thesis proposes integrated modeling and optimization frameworks combining dynamic thermodynamic simulation, sensitivity analysis, and surrogate-based optimization techniques. The proposed methodologies are tested under different operating scenarios characterized by load variability, enabling the evaluation of their robustness and their capability to provide reliable sizing solutions under different operating conditions. From a methodolog-

ical perspective, the thesis introduces a key original contribution: a two-stage sizing workflow that integrates Morris screening with Efficient Global Optimization (EGO) based on adaptively updated Gaussian-process surrogates, fully coupled with the dynamic plant simulation. To the author’s knowledge, this integrated approach has not previously been applied to small-scale CAES systems, where sizing studies generally rely on steady-state formulations, constant-efficiency component models, or exhaustive parametric sweeps.

- **Objective:** To investigate the behavior of small-scale CAES systems under realistic and time-varying operating conditions, with particular emphasis on variable electrical load profiles and pressure-dependent air storage.

**Contribution:** Alongside the development and assessment of the sizing methodology, the thesis investigates the impact of off-design operation on component behavior and overall system performance, providing insights into load-induced performance variations and the effects of sizing choices.

- **Objective:** To assess the technical performance, economic feasibility, and environmental impact of small-scale CAES systems in stand-alone and remote applications integrating renewable energy sources and auxiliary generation units.

**Contribution:** The thesis delivers a comprehensive techno-economic and environmental assessment of hybrid off-grid small-scale CAES systems, combining energy analysis, economic evaluation, and life cycle assessment. A key original contribution is the development of a parametric Life Cycle Inventory framework, in which the material composition of each CAES component is expressed as a function of a characteristic size parameter through correlations derived from extensive market data. This approach enables environmental impacts to be consistently evaluated within the sizing optimization process.

- **Objective:** To analyze the effects of system scaling and component technology selection on the performance of small-scale CAES sys-

tems.

**Contribution:** The thesis provides a detailed modeling and performance assessment of turbomachinery-based CAES configurations, explicitly accounting for the dependence of turbomachinery efficiency on system size and evaluating the resulting impacts on system performance, off-design operation and economic outcomes. A methodological novelty is introduced by enriching map-based compressor and turbine models with a scale-dependent ceiling on the maximum achievable isentropic efficiency and with technological constraints, and in pre-computing the optimal scheduling laws for inlet guide vanes and variable stator vanes through a dedicated off-design optimization, subsequently embedded in the plant model as non-dimensional look-up maps.

- **Objective:** To investigate the potential role of small-scale CAES systems in grid-connected applications under variable electricity market conditions.

**Contribution:** The thesis evaluates price-arbitrage-oriented operating strategies as a function of system size and market dynamics, providing quantitative evidence of the economic potential and limitations of CAES as a complementary storage technology.

By addressing these objectives, this thesis contributes to a more coherent, evidence-based understanding of the realistic potential, limitations, and domain of applicability of small-scale CAES technologies.

## 1.4 Thesis structure

In accordance with the previously described research objectives, this thesis is organized into nine chapters. The subsequent section outlines the content of next chapters of this thesis:

- **Chapter 2** provides a review of the state of the art of energy storage technologies, with particular emphasis on CAES systems. After a general overview of energy storage solutions, the chapter focuses on

CAES technology special attention to small and micro-scale CAES system and their application in off-grid and decentralized contexts.

- **Chapter 3** represents the starting point of the original research activity and introduces the development of a time-dependent thermodynamic model of an existing micro-CAES prototype. This initial system layout is characterized by the availability of experimental data, which enables a thorough validation of the numerical model. An energy and exergy analysis is then carried out to identify the dominant sources of inefficiency and irreversibility in the reference configuration. Overall, the chapter provides the modeling basis for activities addressed in the subsequent chapters.
- **Chapter 4** introduces a sizing methodology based on sensitivity analysis and surrogate-based optimization. A detailed thermodynamic and economic model is presented including CAES integration with renewable generation and electrical loads. The chapter proposes a systematic approach to identify the most influential design parameters and to optimize component sizing under technological and economic objectives.
- **Chapter 5** assesses the robustness and practical relevance of the modeling and sizing methodology developed in the previous chapters by testing it under different operating conditions characterized by different level of load variability. The chapter combines time-dependent thermodynamic simulation, exergy analysis, and techno-economic optimization to quantify how load dynamics affect optimal component sizing, component behavior, and overall system efficiency. The results highlight the importance of scenario-aware and integrated sizing approaches to ensure reliable performance and economic assessments.
- **Chapter 6** focuses on the environmental and economic viability of hybrid micro-CAES systems applied to remote off-grid communities. A combined techno-economic and life cycle assessment framework is



adopted to compare different system configurations under environmental and economic optimization criteria. The analysis provides a holistic evaluation of system performance and environmental impact under real-world conditions.

- **Chapter 7** investigates the scaling of small-scale CAES systems and the integration of radial turbomachinery in off-grid applications. The chapter introduces compressor and radial turbine models, including performance maps, efficiency estimation methods, and control strategies. Particular attention is devoted to the optimization of inlet guide vane and variable stator vane angles under off-design conditions. The proposed models are then integrated into a hybrid off-grid system, where economic assessment and sizing optimization are performed to evaluate the implications of turbomachinery integration at the system level.
- **Chapter 8** extends the analysis to grid-connected small-scale CAES systems, shifting the focus from stand-alone operation to market-oriented applications. A simplified charge–discharge strategy for price-arbitrage operation is introduced, and the economic feasibility of grid-connected CAES is assessed under different electricity price scenarios and incentive mechanisms. This chapter provides insight into the conditions under which small-scale CAES systems may become competitive in interconnected power systems.
- **Chapter 9** summarizes the main findings of the thesis, discusses the overall implications of the results, and outlines possible directions for future research.

## Chapter 2

# State of the art

This chapter reviews the state of the art of energy storage technologies, with the aim of providing the necessary background to position and interpret the work presented in this thesis.

The chapter is structured into three main parts. The first part provides a general overview of energy storage technologies, classifying them according to the primary form of stored energy and discussing their typical applications, maturity levels, and performance characteristics. This section establishes the conceptual framework necessary to contextualize CAES among existing storage options.

The second part focuses on Compressed Air Energy Storage, describing its operating principles and reviewing the main system configurations reported in the literature, including active power plants and existing prototypes.

The third part concentrates on small-scale CAES technologies, which constitute the core focus of this doctoral thesis. The state of the art of small-scale CAES systems is reviewed with particular attention to modeling, sizing methodologies, and life cycle assessment. Key gaps in the literature are identified, and the positioning of the present research within this context is outlined.

## 2.1 Energy storage overview

Energy Storage Systems can be classified according to several criteria, such as storage duration, response time, and application. However, they are most commonly categorized based on the form of energy stored, which provides a clear and comprehensive framework for comparing different technologies. This classification includes the following main categories:

- **Mechanical energy storage** such as Pumped Hydroelectric Energy Storage (PHES), Compressed Air Energy Storage, and Flywheel Energy Storage (FES).
- **Electrochemical energy storage** including Batteries Energy Storage (BES) and Flow Batteries Energy Storage (FBES).
- **Chemical energy storage** such as hydrogen, synthetic natural gas (SNG) and biofuels.
- **Thermal energy storage**, comprising Sensible heat and latent heat Thermal Energy Storage (TES) systems, as well as thermochemical storage based on absorption or adsorption processes.
- **Electrical energy storage**, including conventional capacitors and Supercapacitors Energy Storage (SCES) systems.

### 2.1.1 Mechanical storage systems

#### 2.1.1.1 Pumped Hydro Storage

Pumped hydroelectric energy storage is the most mature and deployed energy storage technology, accounting for approximately 94% of the global installed energy storage capacity [54]. PHES operates by converting electrical energy into gravitational potential energy by pumping water between two reservoirs at different elevations during periods of excess generation, and recovering electricity by releasing the stored water through turbines during high-demand periods [12].

From a performance and economic perspective, PHES combines relatively high cycle efficiencies, typically in the range of 70–85%, with long service lifetimes of 30–60 years [55]. Moreover, its capital cost per unit of installed power is comparatively low, and individual plants can deliver hundreds of megawatts while providing ancillary grid services such as frequency regulation and spinning reserve. Nevertheless, PHES deployment is strongly constrained by geographical and hydrological requirements, as suitable sites require significant elevation differences and reliable water availability [12].

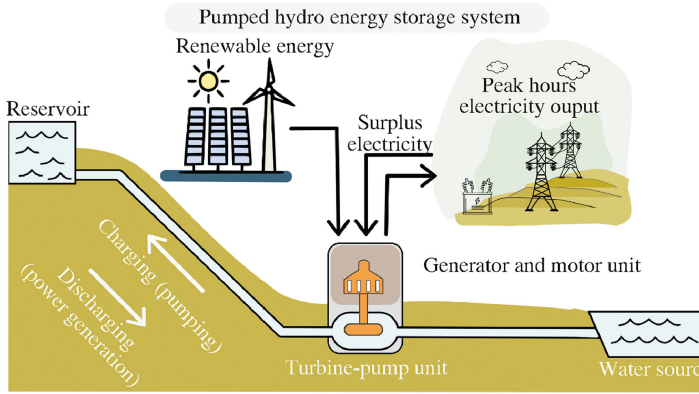


Figure 2.1: Schematic of pumped storage systems[56]

### 2.1.1.2 Compressed Air Energy Storage

Compressed air energy storage enables long-duration energy storage by converting surplus electrical energy into the potential energy of compressed air. During charging, excess electricity drives compressors, while the stored air is subsequently expanded through turbines to regenerate electricity during periods of higher demand [57]. A simplified schematic of a CAES system is shown in Figure 2.2. Within the energy storage landscape, CAES occupies a position similar to PHES while offering greater site flexibility. Compared with electrochemical batteries, CAES

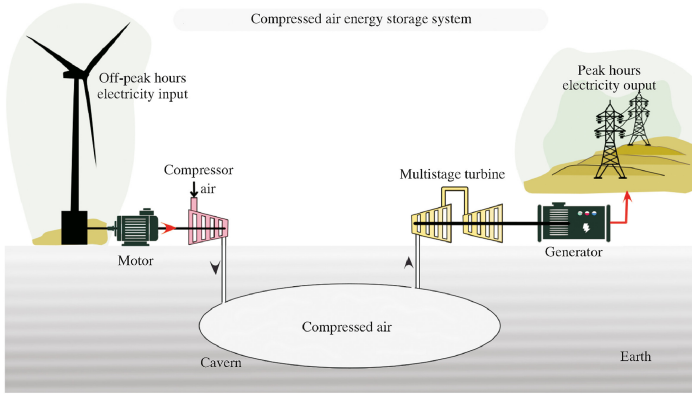


Figure 2.2: Schematic of compressed air energy storage systems [56]

systems typically exhibit slower response times but benefit from large storage capacities, low self-discharge, and long operational lifetimes [58]. Owing to their predominantly mechanical nature, CAES systems rely on mature industrial components and require limited amounts of critical raw materials. These features make CAES suitable for services such as peak shaving, load shifting, seasonal storage and the integration of variable renewable energy sources [22]. Unlike PHES, CAES does not require large water reservoirs or significant elevation differences, but depends on the availability of suitable air storage solutions, including underground geological formations or engineered pressure vessels [59].

From a performance perspective, CAES are characterized by moderate round-trip electrical efficiencies. Conventional utility-scale installations typically achieve efficiencies of approximately 40–55%, while advanced concepts incorporating improved thermal management may approach 60–70%. Small and micro-CAES systems generally exhibit lower efficiencies [27, 58, 60].

### 2.1.1.3 Liquid air energy storage

Liquid air energy storage technology uses surplus electrical energy to liquefy air, store it in insulated cryogenic tanks, and subsequently reheat and expand it to generate electricity during periods of demand [61]. A simplified schematic representation of a LAES system is shown in Figure 2.3. By storing energy in liquid form rather than as compressed gas, LAES achieves significantly higher energy density than CAES [62]. This feature makes LAES largely geographically unconstrained and independent of underground caverns, relying instead on well-established cryogenic technologies and industrial components [63].

Within the energy storage landscape, LAES occupies a role similar to CAES and PHES, while also offering the unique capability to deliver both electrical power and cold energy during discharge [61]. This enables potential integration with industrial processes and recovery of low-grade waste heat. Round-trip electrical efficiencies in the range of approximately 60–70% are considered technically achievable for large-scale LAES [64]. At smaller scales, efficiencies tend to be lower due to the energy intensity of the liquefaction process [61]. Moreover, despite extensive research activity, the large-scale commercial deployment of LAES remains limited, and economic competitiveness relative to alternative long-duration storage

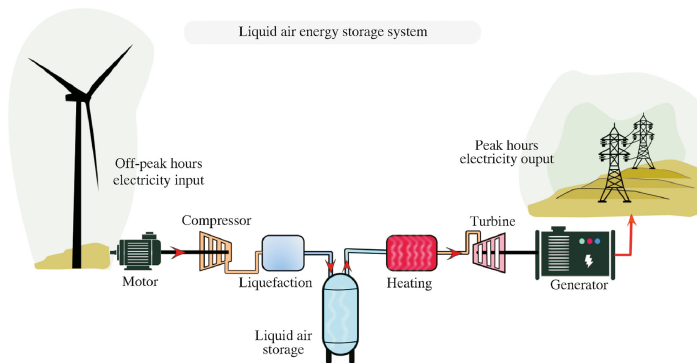


Figure 2.3: Schematic of liquid air energy storage systems [56]

technologies has not yet been demonstrated conclusively [61, 65].

#### 2.1.1.4 Flywheel Energy Storage

Flywheel energy storage systems store energy as rotational kinetic energy in a high-speed rotor housed in a vacuum enclosure and supported by low-friction bearings [66], as shown in Figure 2.4. They exhibit extremely fast dynamic response, delivering rated power within milliseconds, which makes them suitable for short-duration applications such as frequency regulation and mitigation of fast power fluctuations. Flywheels are characterized by short discharge durations, from seconds to a few minutes, very high cycle lifetimes and round-trip efficiencies exceeding 85% [4, 67].

Flywheels are applied in stationary grid-support and transportation systems, enabling rapid capture and release of regenerative braking energy [67]. In power systems, they are often integrated with batteries to provide high-power buffering and mitigate transient loads, despite their higher cost per unit energy compared to bulk storage technologies such as PHES and CAES [4, 12].

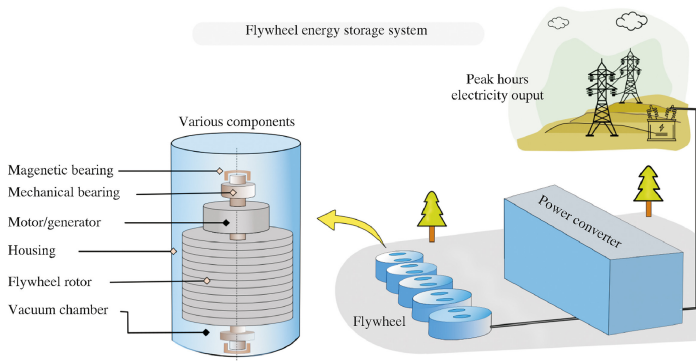


Figure 2.4: Schematic of flywheel energy storage systems [56]

### 2.1.2 Electrochemical Energy Storage Systems

Electrochemical Energy Storage Systems constitute a core class of storage technologies for modern power systems, particularly in microgrid and distributed energy applications. These systems store electrical energy through reversible electrochemical reactions and are characterized by high efficiency, operational flexibility, fast response times, and strong compatibility with renewable energy sources [66, 68, 69].

Electrochemical storage technologies are commonly classified into: (a) battery energy storage (BES) systems, in which energy is stored within the electrodes, and (b) flow battery energy storage (FBES) systems, in which energy is stored in external electrolytes circulated through electrochemical stacks [56].

#### 2.1.2.1 Conventional batteries

Conventional batteries are composed of multiple electrochemical cells, each including an anode, a cathode, and an electrolyte. Depending on the materials employed, battery technologies include lead–acid, lithium-ion, nickel-cadmium, sodium sulphur, sodium-ion, and metal–air systems. Among these, lithium-ion batteries represent the most mature and widely deployed solution for stationary applications. A schematic representation of a lithium-ion battery is shown in Figure 2.5. Lithium-ion batteries dominate residential, commercial, and grid-connected storage due to their

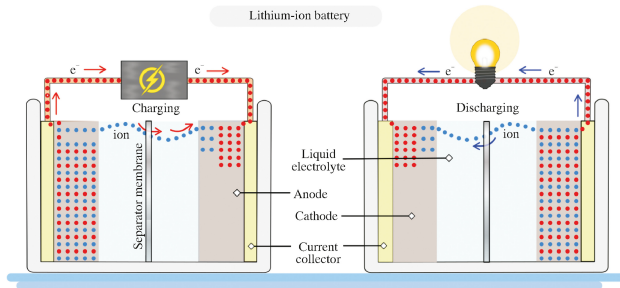


Figure 2.5: Schematic of Li-ion battery [56]



high specific energy, modular scalability, rapid response, and high round-trip efficiency, often exceeding 85–90% under steady operating conditions [68, 70, 71]. These characteristics make them well suited for short to medium-duration applications such as peak shaving, frequency regulation and renewable energy smoothing [66, 69]. However, their long-term performance and economic viability are affected by cyclic and calendric ageing. In addition, they rely on critical raw materials raising concerns related to resource availability and environmental sustainability [13, 72]. Lead–acid batteries remain in use mainly in cost-sensitive applications due to their low upfront cost and technological maturity, but their low energy density and limited cycle life significantly restrict their attractiveness [70].

### 2.1.2.2 Flow batteries

Flow batteries store energy in liquid electrolytes contained in external tanks and circulated through electrochemical cells separated by an ion-selective membrane. A schematic representation of a typical flow battery configuration is shown in Figure 2.6. Major technologies include vanadium redox, zinc–bromine, and polysulfide–bromide systems.

The key advantage of flow batteries lies in the decoupling of energy capacity and power, which allows storage capacity to be scaled indepen-

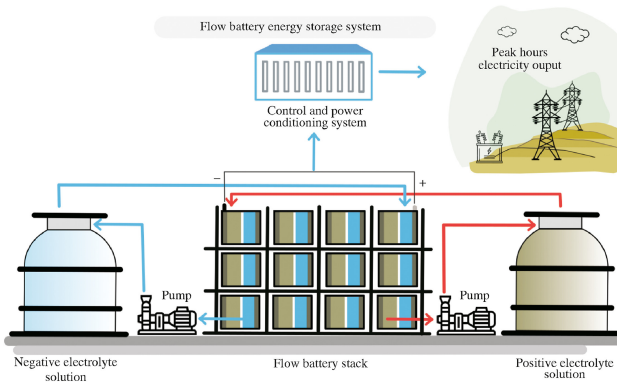


Figure 2.6: Schematic of flow batteries energy storage systems [56]

dently by increasing electrolyte volume [73, 74]. Compared to lithium-ion batteries, flow batteries exhibit lower energy density and larger spatial requirements but offer longer cycle life and reduced sensitivity to deep discharge [66, 68]. Flow batteries are mainly attractive where space constraints are limited and long-duration storage is required.

### 2.1.3 Thermal Energy Storage Systems

Thermal Energy Storage (TES) systems store energy in the form of heat and are commonly classified according to the underlying storage mechanism, namely sensible, latent, and thermochemical storage. While this classification describes the physical principles of heat storage, TES technologies can also play different roles within integrated energy conversion systems, particularly as intermediate storage in power-to-power applications.

#### 2.1.3.1 Classification of Thermal Energy Storage Technologies

**Sensible Heat Storage (SHS)** stores energy by changing the temperature of a storage medium without phase transition. It represents the most mature and widely deployed TES technology due to its simplicity and low cost, with typical media including water, thermal oils, molten salts, and solid materials. However, SHS is characterized by relatively low volumetric energy density, requiring large storage volumes for long-duration applications [58].

**Latent Heat Storage (LHS)** exploits the enthalpy of phase change materials (PCMs), enabling quasi-isothermal operation and higher energy density compared to SHS [58, 75]. Despite these advantages, LHS deployment is often limited by low thermal conductivity and long-term material stability issues [12, 56].

**Thermochemical Energy Storage (TCES)** relies on reversible chemical reactions to store heat, offering the highest theoretical energy density and negligible thermal losses over long storage periods [56, 76]. Nevertheless, its practical application is constrained by system complexity,

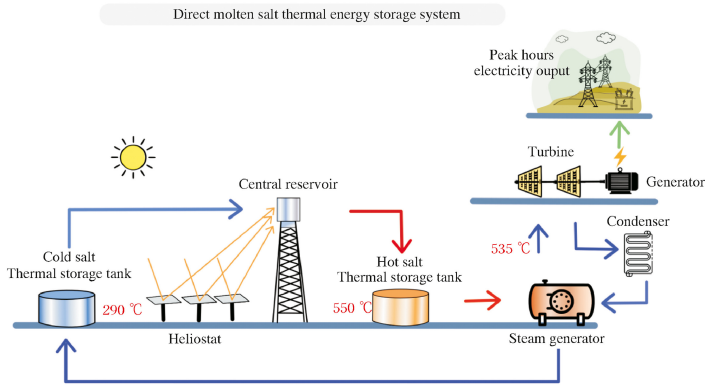
material degradation, and limited technological maturity, confining TCES mainly to research and pilot-scale applications [77].

### 2.1.3.2 Applications of Thermal Energy Storage

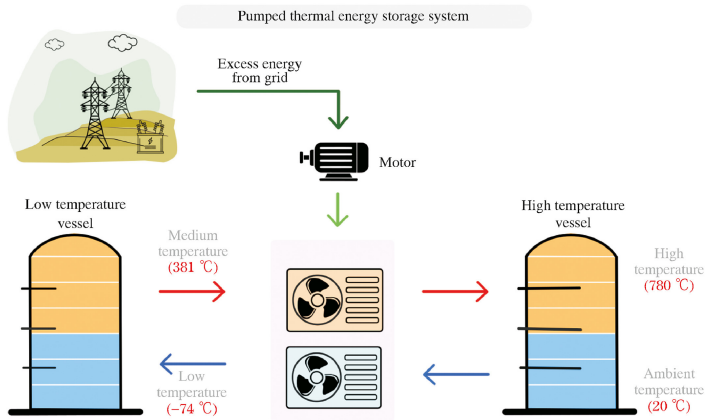
Thermal energy storage can be classified according to the role it plays, distinguishing between thermal-to-thermal applications, where heat is both stored and delivered, and power-to-power applications, in which thermal storage acts as an intermediate medium for electricity storage.

Thermal-to-thermal applications includes space heating and cooling, domestic hot water production, district heating networks, and industrial process heat. Water-based storage dominates low and medium-temperature applications, while molten salts and solid materials are adopted at higher temperatures, particularly in integration with steam Rankine and advanced Brayton cycles [67], like in concentrating solar power (CSP) plants, providing multi-hour storage while also acting as heat transfer fluids [57], as schematically shown in Figure 2.7-a.

Among power-to-power electricity storage systems, Pumped Thermal Energy Storage (PTES) has attracted increasing interest in recent years. PTES systems store electrical energy by converting it into thermal energy through a high-temperature heat pump cycle and subsequently reconvert the stored heat into electricity using a heat engine cycle [56]. Typical layouts include hot and cold thermal reservoirs, turbomachinery operating alternately in compression and expansion modes, and heat exchangers, as illustrated in Figure 2.7-b. Although PTES offers advantages such as reduced geographical dependence and potentially long service life, current implementations exhibit relatively low round-trip electrical efficiencies, typically in the range of 40–50%, and therefore remain largely confined to pilot and demonstration-scale projects [78, 79].



(a) Thermal energy storage integration in a concentrating solar power plant.



(b) Schematic representation of a pumped thermal energy storage system.

Figure 2.7: Representative applications of thermal energy storage. [56]

### 2.1.4 Chemical energy storage

Chemical Energy Storage systems store energy in the chemical bonds of molecules and are particularly suited for long-term applications. Key CES technologies include hydrogen energy storage and synthetic fuels. Among

these, hydrogen has received increasing attention due to its potential role in deep renewable integration and its carbon-free nature [56, 80].

#### 2.1.4.1 Hydrogen energy storage

Hydrogen can be produced from water via electrolysis or solar-driven processes and stored for later use as a chemical energy carrier. A typical hydrogen energy storage system consists of a hydrogen production unit, a storage subsystem, and an energy conversion device, such as a fuel cell, to reconvert the stored chemical energy into electricity [55], as schematically shown in Figure 2.8.

Hydrogen storage is particularly suitable for large-scale and long-duration applications, due to its low self-discharge [81]. However, the multiple energy conversion steps involved introduce efficiency penalties and system complexity. Consequently, hydrogen energy storage systems typically exhibit relatively low round-trip electrical efficiencies, commonly in the range of 25–45%, depending on the electrolyser and fuel cell technologies [82]. Hydrogen storage also faces significant technical challenges related to its low volumetric energy density, which requires compressed, liquefied, solid-state, or underground storage solutions, and to material compatibility issues such as hydrogen embrittlement [81, 82]. Furthermore,

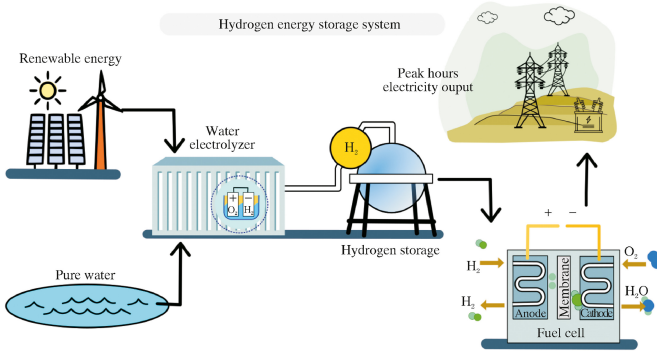


Figure 2.8: Schematic of hydrogen based energy storage systems [56]

the environmental benefits of hydrogen energy storage strongly depend on the electricity source used for hydrogen production [80]. Despite these limitations, hydrogen energy storage remains a major focus of research and demonstration activities, with future deployment strongly dependent on technological maturation.

### 2.1.5 Electrical energy storage systems

Electrical energy storage technologies directly store electricity without intermediate conversion and are characterized by very fast response times and high efficiency.

Among these technologies, supercapacitors are well established for high-power, short-duration applications. They store energy electrostatically at the electrode–electrolyte interface [66] and, unlike electrochemical batteries, do not rely on bulk chemical reactions, enabling extremely rapid charge–discharge cycles and exceptional cycling durability [68]. As a result, supercapacitors achieve very high specific power, round-trip efficiencies typically up to 95%, and response times on the order of milliseconds [4]. Due to their characteristics, supercapacitors are not suitable for bulk energy storage but excel in short-duration applications such as voltage support, frequency regulation, and transient smoothing. They are often deployed in hybrid configurations with higher-capacity storage technologies to absorb fast transients, finding application in regenerative braking, uninterruptible power supply buffering, and grid stabilization services requiring sub-second response [4, 58, 66].

### 2.1.6 Comparative framework for energy storage technologies

The energy storage technologies reviewed in Sections 2.1.1 to 2.1.5 span several orders of magnitude in rated power, energy capacity, discharge duration, and lifetime. As a consequence, a meaningful comparison cannot be based on a single performance indicator but requires a system-level perspective that accounts for architectural constraints and application-specific requirements. A quantitative overview of representative performance values commonly reported in the literature is provided in Table 2.1, while Figure 2.9 illustrates the broader range of potential application domains in the power–energy plane.

A first and fundamental axis of differentiation is the temporal scale of storage. Technologies such as supercapacitors and flywheels are characterized by extremely high power density and fast response times, but limited energy capacity, enabling discharge durations from seconds to minutes. These features make them inherently suitable for power-quality services and short-term balancing. At the opposite end of the spectrum, large-scale mechanical storage systems, including PHES and CAES, operate at lower power density but can sustain discharge over several hours or even days, enabling bulk energy shifting and long-duration balancing. Between these extremes lies a broad continuum in which trade-offs become strongly application-dependent. In this intermediate range, electrochemical batteries, particularly lithium-ion, currently represent the dominant solution, combining high round-trip efficiency, fast response, and technological maturity. Their modularity and declining costs have enabled widespread deployment across residential, commercial, and grid-scale applications. However, their suitability progressively decreases as storage duration and cumulative throughput increase, due to finite lifetime, sensitivity to cycling depth, and reliance on critical raw materials. Beyond performance metrics alone, the comparison highlights a strong link between scalability and system architecture. As shown in Figure 2.9, battery-based systems tend to align along the power–energy bisector, reflecting the intrinsic coupling between power and energy capacity in modular electrochemical

designs. In contrast, storage concepts based on reservoirs, pressure vessels, or geological formations, such as PHES and CAES, exhibit a wider spread in the power–energy plane. In these systems, power is primarily determined by turbomachinery, while energy capacity depends on storage volume, enabling a partial decoupling of power and capacity and greater flexibility in system design at large energy scales. Thermal and chemical storage technologies further enrich the storage landscape, although their effectiveness depends strongly on system integration. High efficiencies can be achieved when heat is stored and recovered within thermal networks, whereas power-to-power configurations are typically penalized by additional conversion stages and associated efficiency losses.

Overall, this comparison confirms that no single energy storage technology simultaneously maximizes efficiency, energy density, power capability, lifetime, and siting flexibility. Instead, the storage landscape is inherently heterogeneous, reinforcing the need for application-specific technology selection and, increasingly, for hybrid architectures combining complementary storage solutions. Within this framework, Compressed Air Energy Storage emerges as a particularly relevant candidate for medium to long-duration applications, offering favorable scalability, long service life, and reduced material intensity compared to electrochemical batteries, while avoiding the geographical constraints of pumped hydro. These characteristics motivate a more detailed examination of CAES technologies, their architectures, and their performance.



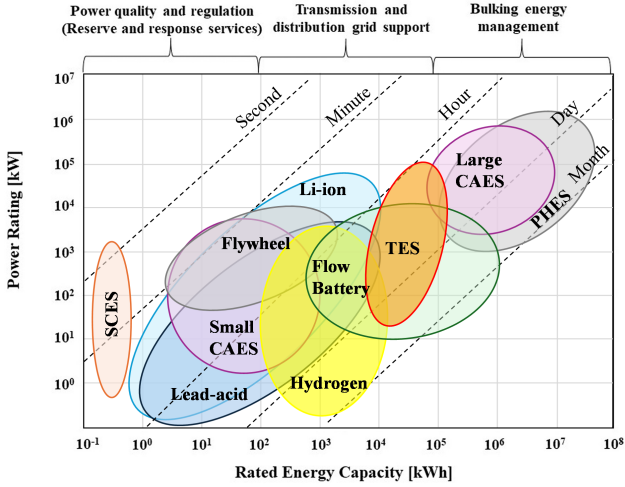


Figure 2.9: Comparison of energy storage systems based on their rated power and capacity (adapted from [27])

Table 2.1: Comparative parameters across major energy storage technologies

| Technology | Rated Power (MW) | Efficiency (%) | Storage Duration | Discharge Time | Life Cycle (yr) | Energy Density (Wh/m <sup>3</sup> ) | Power Density (kW/m <sup>3</sup> ) |
|------------|------------------|----------------|------------------|----------------|-----------------|-------------------------------------|------------------------------------|
| PHES       | 10–5000          | 70–85          | h–month          | 1–24 h         | 30–60           | 0.5–2                               | 0.1–1.5                            |
| CAES       | 100–1000         | 40–75          | h–month          | 1–24 h         | 20–50           | 2–6                                 | 0.5–2                              |
| Small CAES | 0.003–10         | 20–50          | h–day            | min–h          | 20–40           | >6                                  | >2                                 |
| LAES       | 0.1–200          | 50–70          | h–month          | 1–24 h         | 25+             | 60–200                              | 200–1000                           |
| Flywheel   | 0.1–20           | 70–95          | sec–min          | sec–min        | 15–20           | 20–400                              | 1000–2000                          |
| Li-ion     | 0–10             | 85–98          | min–day          | min–h          | 5–15            | 200–500                             | 100–10000                          |
| Lead-acid  | 0.1–20           | 70–90          | min–day          | sec–h          | 5–15            | 50–80                               | 90–700                             |
| VRB        | <10              | 70–85          | h–month          | sec–h          | 5–10            | 20–70                               | 0.5–2                              |
| Hydrogen   | 0.1–50           | 20–50          | h–month          | sec–24 h       | 10–20           | 500–3000                            | >500                               |
| SCES       | 0–0.3            | 75–95          | sec–min          | millisec–h     | >20             | 10–30                               | 40000–120000                       |
| PTES       | 10–150           | 40–60          | h–day            | h–day          | 15–20           | 10–100                              | 1–15                               |
| LHS        | 0–250            | 50–90          | min–month        | 1–24 h         | 10–30           | 100–370                             | 150–250                            |
| SHS        | 0–5              | 75–90          | min–month        | 1–24 h         | 10–31           | 25–120                              | 10–12                              |

## 2.2 Overview of compressed air energy storage

The general working principle of CAES, as described in Section 2.1.1.2, is based on the conversion of surplus electrical energy into the potential energy of compressed air. From a functional standpoint, a CAES system comprises four main subsystems: compression, air storage, thermal management, and expansion with power conversion.

Compressed air is stored in a dedicated reservoir whose characteristics strongly influence system design and implementation. Large-scale CAES installations typically rely on underground geological formations, such as salt caverns or depleted hydrocarbon reservoirs, which enable operation at high pressures with low leakage rates [58, 59]. In small-scale or decentralized configurations, storage is commonly achieved using engineered pressure vessels, leading to different constraints in terms of pressure range, energy density, and integration with downstream components [59]. Although CAES systems may also be distinguished based on the type of air storage employed, this aspect is only marginally treated.

Although conceptually simple, overall system performance is governed by the strong coupling between mechanical work and thermal effects. Air compression is inherently accompanied by heat generation, while expansion results in significant cooling. During the charging phase, the temperature rise associated with compression affects component sizing, operating limits, and efficiency, whereas during discharge the temperature drop can adversely impact expander performance and reliability. Overall cycle efficiency therefore depends not only on the intrinsic performance of compressors and expanders, but also on how effectively thermal effects are managed throughout the cycle [34].

Within this functional framework, CAES architectures can be distinguished primarily by the treatment of thermal energy during compression and expansion. On this basis, CAES systems are commonly classified into diabatic, adiabatic, and isothermal configurations [59], which are discussed in detail in the following section.

## 2.2.1 Classification of CAES architectures

### 2.2.1.1 Diabatic CAES

Diabatic compressed air energy storage (D-CAES) is the earliest and historically established CAES configuration, characterised by the rejection of the heat generated during air compression rather than its storage [21]. During the charging phase, compression heat is dissipated through intercoolers before the air is stored in pressurised tanks or geological reservoirs. During discharge, the compressed air must be reheated using an external energy source, typically through fuel combustion, to ensure suitable turbine inlet temperatures [12]. This operating principle classifies D-CAES as a hybrid system combining mechanical energy storage with fuel-based electricity generation. A schematic layout of a D-CAES system is shown in Figure 2.10. From a thermodynamic perspective, rejecting compression heat leads to significant exergy losses and relatively low round-trip efficiencies, commonly reported in the range of 40–55% [12]. Although the absence of thermal energy storage (TES) reduces system complexity and initial capital costs, this advantage is counterbalanced by increased reliance on external heating, higher operational costs, and associated emissions.

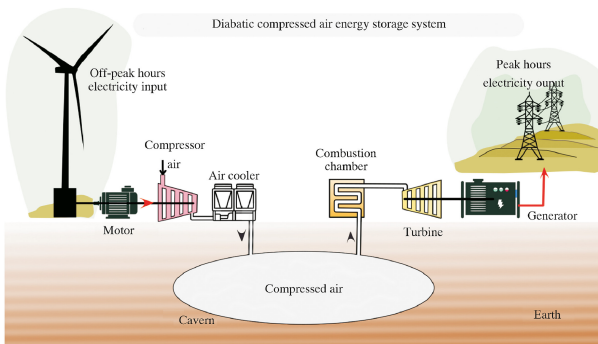


Figure 2.10: Schematic representation of diabatic compressed air energy storage (D-CAES) plant [56]

D-CAES remains the only CAES configuration with commercial-scale operational plants, namely the Huntorf and McIntosh facilities. However, its limited efficiency and dependence on external reheating have constrained further deployment. The use of recuperators to recover exhaust heat can partially improve thermal performance, but overall efficiency remains insufficient to meet current market requirements [34]. Moreover, fuel combustion reintroduces greenhouse gas emissions into systems otherwise promoted for their sustainability. While alternative fuels such as hydrogen can reduce emissions at the point of use, upstream production pathways may offset these benefits when assessed from a full lifecycle perspective [21].

#### 2.2.1.2 Adiabatic CAES

Adiabatic compressed air energy storage (A-CAES) differs fundamentally from diabatic configurations in that the heat generated during air compression is retained and stored for later reuse during the expansion phase, thereby eliminating the need for external fuel-based reheating [60].

In principle, an adiabatic system could preserve the compression heat directly within the stored air. However, direct storage of hot compressed air, while conceptually straightforward, poses significant technical challenges. These include stringent requirements on component materials and lubrication systems, reduced compressor efficiency at elevated temperatures, and severe thermal constraints on storage vessels. As a result, such configurations have not found practical application.

To overcome these limitations, adiabatic CAES systems typically integrate a dedicated thermal energy storage (TES) unit, which absorbs the heat removed after each compression stage and releases it during the expansion process [34]. This approach eliminates fuel combustion, allows air to be stored at lower temperatures and higher pressures, and removes constraints associated with the thermal resistance of storage reservoirs. These advantages are achieved at the expense of increased system complexity due to the inclusion of additional heat exchangers and thermal storage components.

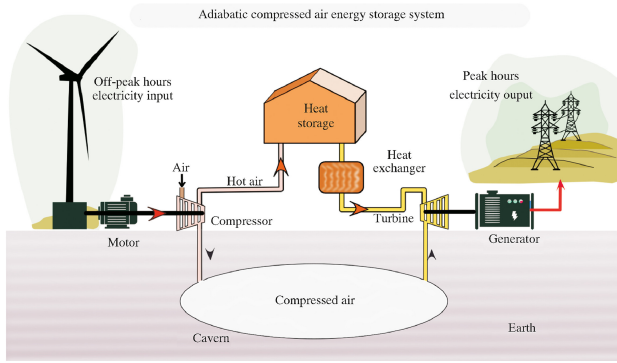


Figure 2.11: Schematic representation of adiabatic compressed air energy storage (A-CAES) plant [56]

The resulting closed thermal loop enables elevated turbine inlet temperatures without combustion, leading to reduced operational costs and zero direct emissions. Systems employing this enhanced thermal integration are commonly referred to as Advanced Adiabatic CAES (AA-CAES). Typical TES storage media include packed-bed solids, molten salts, mineral oils, and pressurized water [34]. A schematic layout of an A-CAES system with integrated TES is shown in Figure 2.11.

By internally recycling compression heat, A-CAES significantly reduces exergy losses associated with heat rejection and achieves higher round-trip efficiencies, which may exceed 70% under optimal operating conditions [83].

### 2.2.1.3 Isothermal CAES

I-CAES consists of realising compression and expansion process as isothermal. While in adiabatic configurations heat is removed or released before compression or after expansion, in I-CAES heat transfer occurs constantly during transformations [84]. Several methods have been investigated to achieve near-isothermal processes, and they can be divided into direct and indirect heat transfer. As it regards direct methodologies,

the most common configurations involve the injection of water droplets into the compressor: the heat from compression is absorbed to reduce the temperature increase, while additional heat is added during expansion to maintain temperature control, another deeply studied approach is to use liquid pistons [85, 86]. For indirect heat transfer techniques, the only way to obtain isothermal processes is by improving heat transfer methods, and it can be accomplished, for example, using finned pistons, porous media, or metal plate inserts. All existing prototypes are based on the use of reciprocating machines, which can perform a slower compression or expansion compared to dynamic ones, and so increase heat transfer[87]. A schematic layout of an I-CAES system implementing both liquid piston and water spray is shown in Figure 2.12. By operating with quasi-isothermal transformations, the final temperature after compression remains relatively constant. Therefore, TES is not required, nor is combustion, eliminating the problems associated with high temperatures and pollutant emissions at the same time. Additionally, the power required for the compression phase is much lower, and the theoretical efficiency is the highest among CAES technologies, making this technology promising. However, I-CAES is still mainly at a research stage [88].

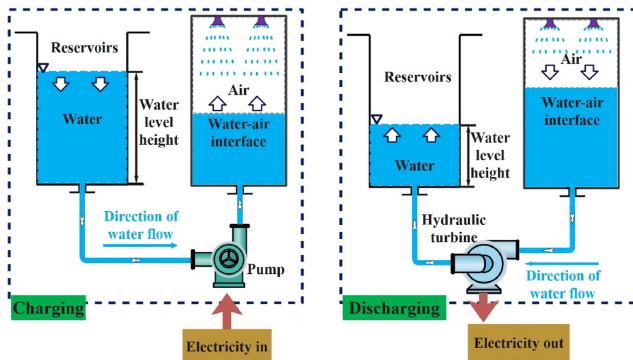


Figure 2.12: Schematic representation of isothermal compressed air energy storage (I-CAES) [89]

### 2.2.2 Thermodynamic comparison of CAES configuration

This section report a thermodynamic comparison of this three principal CAES configurations, comparing the corresponding  $p-v$  and  $T-s$  diagrams together with the simplified process layouts. The diagrams are intended for illustrative purposes only and are not drawn to scale; their role is to convey the qualitative differences in the thermodynamic paths followed by the working fluid during charging and discharging, rather than to reproduce quantitatively realistic state points.

In the  $p-v$  plane, where the area subtended by each curve corresponds to the specific work of compression and expansion, the three configurations exhibit marked differences. In the diabatic case (Figure 2.14b), the expansion work is appreciably larger than the compression work, the difference being supplied by the combustion chamber through fuel addition.

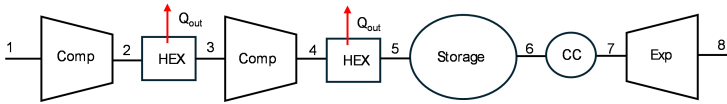


Figure 2.13: Simplified D-CAES block diagram

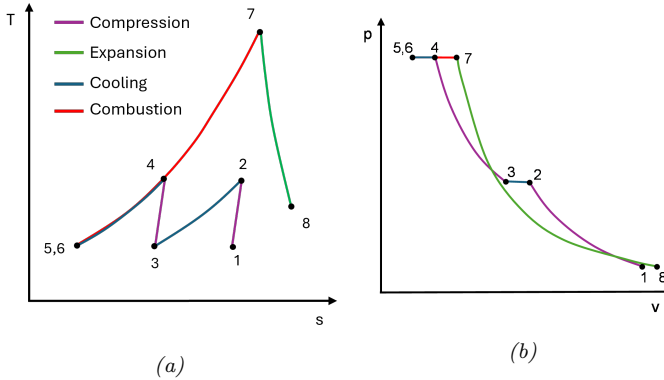


Figure 2.14: Thermodynamic diagram for D-CAES: (a) Temperature-entropy (b) Pressure-volume

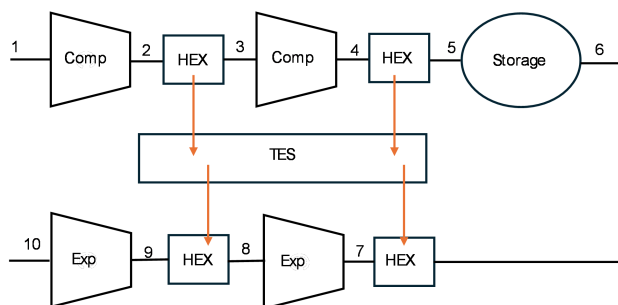


Figure 2.15: Simplified A-CAES block diagram

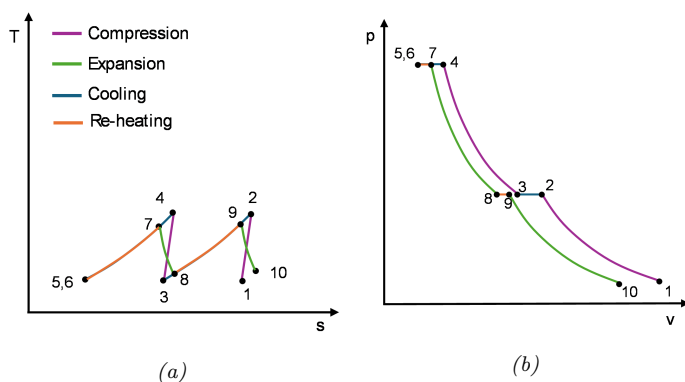


Figure 2.16: Thermodynamic diagram for A-CAES: (a) Temperature-entropy (b) Pressure-volume



Figure 2.17: Simplified I-CAES block diagram



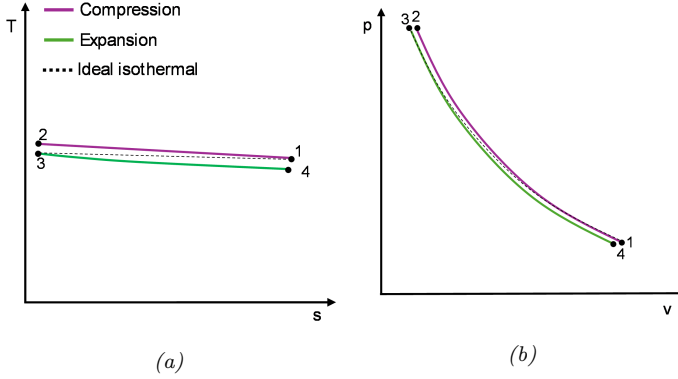


Figure 2.18: Thermodynamic diagram for I-CAES: (a) Temperature-entropy (b) Pressure-volume

In the adiabatic configuration (Figure 2.16b), by contrast, the work delivered during the expansion is lower than the work absorbed during the compression phase, since no external thermal input compensates for the unavoidable irreversibilities and for the finite effectiveness of the thermal energy storage employed to recover the compression heat. The isothermal configuration (Figure 2.18b) approaches the ideal limit in which the compression and expansion paths collapse onto a common isotherm and the two areas become essentially equal: the closer the actual transformation lies to a true isotherm, the smaller the gap between the two curves. The accompanying  $T-s$  diagrams (Figures 2.14a, 2.16a and 2.18a) highlight the temperature range spanned by each configuration. In the diabatic plant, the peak cycle temperature is substantially higher than in the other two layouts, since combustion in the dedicated combustor raises the gas temperature well above what is attainable by compression alone, the upper bound being set in practice by the turbine inlet temperature limits of the expander. In the adiabatic case, heat is extracted between compression stages and transferred to the TES, which simultaneously reduces the compression work and keeps the peak temperature within the limits tolerated by the TES medium and the associated heat exchangers;

this value typically lies well below diabatic combustion temperatures. The isothermal configuration deliberately suppresses temperature excursions altogether: heat is rejected during compression and supplied during expansion at a rate sufficient to keep the working fluid close to ambient conditions, so that the entire cycle is confined to a narrow band in the  $T$ - $s$  plane. The progressive flattening of the cycle, moving from D-CAES to A-CAES and finally to I-CAES, brings the compression and expansion processes closer to the ideal isothermal transformation, which, for a given pressure ratio, minimises the compression work and maximises the expansion work.

### 2.2.3 Historical Development and Current Landscape of CAES Projects

Historically, CAES development has been shaped by a small number of reference plants and a wider set of proposed, delayed, or discontinued projects. This experience illustrates both the technical feasibility of CAES and constraints on its large-scale commercialization [60].

The most influential benchmark is the Huntorf plant in northern Germany, commissioned in 1978 as the first commercial CAES facility. Initially rated at 290 MW and later refurbished to 321 MW in 2006, Huntorf operates with a diabatic CAES cycle and stores compressed air in deep salt caverns [60]. Although its round-trip efficiency is relatively low (around 42%), its decades-long operation demonstrates the mechanical robustness of CAES and the suitability of salt caverns for repeated pressure cycling [90]. The second key reference is the McIntosh plant in Alabama (USA), commissioned in 1991 with a capacity of 110 MW. Also based on a diabatic configuration with underground cavern storage and fuel-based reheating, McIntosh incorporates improved recuperation and is typically associated with higher efficiencies of about 54% [21]. Together, Huntorf and McIntosh remain the primary operational proof points for utility-scale CAES technology [21, 60].

Beyond these successful deployments, several proposed projects highlight both the ambition to scale D-CAES technologies and the challenges

associated with their realization. The Norton Energy Storage project in Ohio (USA) envisioned a 2700 MW D-CAES facility based on a repurposed limestone mine, with the objective of coupling CAES with renewable energy sources to mitigate their inherent intermittency. Despite its scale and strategic intent, the project was terminated in 2013 due to insufficient progress by the developer [91]. Similarly, the Iowa CAES project, proposed in 2003, planned a comparably sized 2700 MW system utilizing a sandstone aquifer as the storage reservoir. This project was discontinued in 2011, underscoring the heightened uncertainty and risk associated with non-salt geological storage media [21, 92].

### 2.2.3.1 From diabatic to adiabatic and isothermal

Recent developments in compressed air energy storage have been driven by the need to improve round-trip efficiency and to eliminate the reliance on fossil fuels that characterizes conventional diabatic systems. Consequently, both research and deployment activities have progressively shifted toward adiabatic and near-isothermal concepts.

Pilot and prototype A-CAES systems have demonstrated RTE exceeding 70% at small scale when high-temperature thermal energy storage and well-integrated heat exchangers are employed, highlighting the central role of thermal integration in system performance [59].

The first large-scale A-CAES project, ADELE project, was developed in Germany by a consortium of several research and industrial partners. The project targeted a round-trip efficiency of approximately 70%, with a design capacity of 90 MW and 360 MWh of stored energy, primarily aimed at facilitating large-scale wind power integration. Despite its technological ambition, the ADELE project was placed on hold in 2016 due to unresolved challenges related to high-temperature compressor development, TES material durability, and uncertain market conditions [35]. In the same year, the ALA CAES Group, in collaboration with the Swiss Federal Office, successfully constructed and tested a 1 MW pilot plant. Commissioned in 2016, the facility uses a sealed transportation tunnel for underground air storage and reported round-trip efficiencies between

63% and 74%, demonstrating the feasibility of alternative underground storage infrastructures. [21]

China has since emerged as a major driver of A-CAES development. The TICC-500 pilot plant (0.5 MW-0.5MWh), commissioned in 2014, was the world's first A-CAES system integrating thermal energy storage. While the plant provided valuable operational insights, it also revealed key thermal management challenges, underscoring the need for robust heat control strategies [21]. Building on this experience, China commissioned a 10 MW-40 MWh integrated A-CAES in Bijie, Guizhou Province. This system employs pressure pipelines for air storage and achieves a reported round-trip efficiency of 60.2%. At the commercial scale, a 50 MW-300 MWh salt-cavern-based CAES power plant has been constructed and commissioned in Feicheng, Shandong Province, achieving an efficiency of approximately 60%. Several additional large-scale projects are currently under development in China. In Jintan, Jiangsu Province, a 50 MW-200 MWh A-CAES plant is already operational, with long-term expansion plans that exceed 1 GW of installed capacity. In Zhangjiakou, Hebei Province, a 100 MW-400 MWh facility utilizing pressure vessels for air storage is presently under construction. [93]

Outside Asia, Hydrostor has developed a distinctive adiabatic CAES concept based on hydrostatically compensated air storage enabling near-constant pressure operation. Its commercial reference plant in Goderich, Canada (1.75 MW-10 MWh), operational since 2019. The system captures and reinjects compression heat, achieving a round-trip efficiency of approximately 60% [21].

Near-isothermal concepts constitute an additional pathway aimed at improving system efficiency; however, significant scaling challenges persist [84]. One of the few recent demonstrations of an isothermal CAES system was developed by SustainX, which designed, built, and tested a 1.5 MW prototype. The system was based on the compression and expansion of a foam-air mixture, enabling enhanced heat transfer and thus near-isothermal operation [94]. The prototype achieved a round-trip efficiency of approximately 54%. A key limitation to the scalability of

this approach was the reliance on specially constructed above-ground pressure vessels for air storage, which constrained both system size and economic viability. Following the prototype demonstration, SustainX divested on this research line [21]. Numerous other near-isothermal CAES concepts have been investigated; however, these efforts largely remain at the prototypal stage and are typically limited to the development of individual components rather than fully integrated systems [91].

Overall, the evolution from diabatic toward adiabatic and isothermal CAES reflects a clear trend toward higher efficiency, zero-fuel operation, and broader geological applicability. While several projects remain at the pilot or demonstration stage, ongoing construction activities and planned large-scale deployments indicate growing technological maturity and increasing confidence in advanced CAES as a long-duration energy storage solution. A summary of the main CAES projects discussed above is provided in Table 2.2.

*Table 2.2: Summary of representative CAES projects and initiatives*

| Project        | Location    | Technology | Storage medium                     | Power (MW) | RTE (%)    | Status             |
|----------------|-------------|------------|------------------------------------|------------|------------|--------------------|
| Huntorf        | Germany     | D-CAES     | Salt cavern                        | 290–321    | 42         | Operational        |
| McIntosh       | USA         | D-CAES     | Salt cavern                        | 110        | 54         | Operational        |
| Norton         | USA         | D-CAES     | Limestone mine                     | 2700       | –          | Terminated         |
| Iowa           | USA         | D-CAES     | Sandstone aquifer                  | 2700       | –          | Discontinued       |
| ADELE          | Germany     | A-CAES     | salt cavern                        | 90         | 70         | Terminated         |
| ALA CAES       | Switzerland | A-CAES     | Sealed transportation tunnel       | 1          | 63–74      | Pilot              |
| TICC-500       | China       | A-CAES     | Pressure vessel                    | 0.5        | 33.3–41.03 | Pilot              |
| Bijie          | China       | A-CAES     | Pressure pipelines                 | 10         | 60.2       | Operational        |
| Feicheng       | China       | A-CAES     | Salt cavern                        | 50         | 60         | Commissioned       |
| Jiangsu Jintan | China       | A-CAES     | Salt cavern                        | 50         | 58.2       | Operational        |
| Goderich       | Canada      | A-CAES     | Hydrostatically compensated cavern | 1.75       | 60         | Operational        |
| Zhangjiakou    | China       | A-CAES     | Pressure vessels                   | 100        | 70.2       | Under construction |
| SustainX       | USA         | I-CAES     | Pressure vessels                   | 1.5        | 54         | Discontinued       |

#### 2.2.4 Research trends and emerging application domains of CAES

The overview presented in Section 2.2.3 focused on CAES plants that have reached, or at least attempted to reach, a commercial and

operational stage. Contemporary CAES research aims to address the limitations that have constrained CAES deployment (i.e. moderate round-trip efficiency, thermal irreversibilities, and strong site dependency) or to explore alternative application domains. The current body of literature can be interpreted through a set of converging research trends.

A first research trend in the CAES literature concerns advanced thermal management strategies aimed at improving round-trip efficiency and reducing exergy losses. This part of literature reflects the transition from conventional diabatic layouts toward thermally integrated configurations, with a sustained focus on how compression heat can be effectively recovered and reutilized during expansion. Within this context, the literature shows growing interest in innovative TES concepts, such as phase-change materials and thermochemical storage systems [95–97], as well as in improved methodologies for TES sizing, integration, and control [98]. Together, these efforts reflect a mature yet still active research trajectory focused on consolidating and optimizing thermally integrated A-CAES architectures.

A second major research trend explores near-isothermal CAES concepts, with a growing focus on the practical limits of heat transfer enhancement rather than on idealized thermodynamic performance. Experimental and numerical studies show that liquid-assisted compression and expansion, can effectively limit temperature variations and reduce compression work at small scale [86, 99, 100]. However, near-isothermal operation strongly depends on long compression times, large heat exchange surfaces, and intense liquid–gas interaction, which favor low-speed reciprocating architectures and constrain achievable power density and scalability [84, 86]. In addition, the complex transient flow behavior inside liquid piston devices poses challenges for thermal control under variable operating conditions [100]. Consequently, recent research increasingly addresses system integration and hybrid configurations, suggesting that near-isothermal CAES is currently more suitable for small-scale or distributed applications rather than utility-scale plants [84, 101].

A third research direction focuses on siting flexibility and alterna-

tive storage solutions in order to overcome the geological constraints traditionally associated with CAES. Alongside salt caverns and aquifers, recent studies and pilot projects investigate mined rock caverns, depleted reservoirs and underwater membrane-based storage [102–104]. This diversification of storage options directly supports the extension of CAES into regions and applications where conventional cavern-based solutions are impractical, enabling greater modularity and scalability.

A fourth trend emphasizes system-level integration and multi-energy functionality. Rather than operating solely as bulk electricity storage assets, modern CAES systems are increasingly studied as components of integrated energy systems. This includes coupling with renewable generation for curtailment reduction and peak shifting [32, 105], as well as exploiting compression heat and expansion cold for combined electricity, heating, and cooling applications [28, 30]. Such polygeneration configurations improve overall system utilization and economic viability, particularly in distributed contexts where local thermal demands coexist with electrical loads.

Within this broader research landscape, a rapidly expanding trend is the investigation of CAES for decentralized and small-scale energy systems [105–107]. The ongoing decentralization of power systems, the diffusion of distributed renewable generation, and the need for resilient long-duration storage in microgrids and isolated networks motivate solutions that are robust, durable, and less dependent on critical materials. In this context, small-scale CAES systems, emerge as attractive candidates due to their mechanical longevity, high cycling capability, and compatibility with multi-energy integration [18, 19].

However, downsizing CAES is not a straightforward scaling exercise. Small-scale systems typically rely on above-ground pressure vessels, resulting in isochoric storage and continuously varying pressure levels during charge and discharge. As a consequence, compressors and expanders operate predominantly under off-design conditions, while thermal losses become comparatively more significant [26, 43]. Moreover, decentralized applications are characterized by strongly time-varying operating profiles

driven by stochastic renewable generation and local demand fluctuations, which further increase the relevance of dynamic behaviour and advanced control strategies [108]. These features clearly distinguish small-scale CAES from utility-scale installations and justify the need for a dedicated analysis, while also motivating continued research and development efforts aimed at fully assessing and unlocking the potential of this technology within future decentralized energy systems.

## **2.3 Small-Scale CAES: Architectures, performance and applications**

Within the range of applicability of CAES technology, the literature does not provide a unique or universally accepted definition of small-scale systems, and different power-based classifications are commonly adopted. Based on recent review works [26, 27], micro-CAES systems are typically defined as installations with rated power below 1 MW, while the term small-scale CAES generally refers to systems with power capacities below 10 MW. Large-scale CAES installations, by contrast, are usually characterized by rated output powers exceeding 50 MW. In this work, the analysis follows this classification framework. All investigated systems fall within the small-scale CAES range, with rated output powers below 10 MW, and particular emphasis is placed on micro-CAES applications with power capacities lower than 1 MW.

As discussed in the previous section, the majority of existing CAES installations are developed at large scale. Although several prototype systems have been realized at smaller scales, and their nominal size may fall within the micro or small-CAES range, their primary objective is not direct small-scale deployment. Rather, these installations are conceived as pilot plants aimed at investigating technological solutions and operational concepts intended to be subsequently scaled up to utility-scale applications. In contrast, experimental analysis of CAES systems specifically designed for direct small-scale applications remain scarce. Their presence in the



literature is very limited and is mainly confined to laboratory-scale setups or component-level experimental rigs, rather than fully integrated system demonstrators [44, 109]. Accordingly, the majority of published research on small-scale CAES is focused on theoretical and modeling-based studies, with comparatively few contributions addressing experimental validation at the system level, as will be detailed later.

### 2.3.1 Architectures and performance of small-scale CAES

The state of the art of micro and small-scale Compressed Air Energy Storage (CAES) systems is characterized by a wide variety of architectures, components, and integration strategies.

According to the classification already discussed in the previous section, small CAES architectures can be grouped into diabatic, adiabatic, and isothermal configurations. Regarding diabatic configurations, while they may be attractive at large scale due to favorable cost-to-efficiency ratios, in small-CAES applications the required fuel handling and combustion infrastructure tends to become disproportionately complex relative to the achievable energy output. Furthermore, diabatic CAES systems generally exhibit low round-trip efficiency at small scale, where thermal losses play a dominant role in overall system performance [35]. Adiabatic configurations are more extensively addressed in the literature, as they represent a compromise between efficiency improvement and technological feasibility. In small-scale applications, TES is most commonly implemented using packed-bed or liquid sensible heat storage units coupled with intermediate heat exchangers. The literature consistently identifies TES design and integration as critical at small scale, affecting both cost and achievable thermal effectiveness, while high-temperature operation introduces increasingly stringent material and design constraints due to unfavorable surface-to-volume ratios [26, 27]. Quasi-isothermal CAES concepts have been proposed as an alternative strategy to mitigate compression and expansion losses by enhancing heat transfer throughout the cycle. However, existing studies predominantly focus on the analysis and development of individual components, while system-level demonstrations

and integrated assessments remain limited [84].

Among these three categories, the literature places a stronger emphasis on adiabatic configurations, which are generally regarded as the most promising compromise between efficiency improvement and technological maturity at small scale [26]. Consequently, adiabatic configuration is adopted as the primary reference in the rest of this thesis.

Several examples of purely adiabatic micro-CAES systems can be found in the literature, primarily investigated through numerical simulation. Dib et al. [110] developed an analytical model of a 20 kW micro-CAES system, reporting an overall system efficiency of approximately 30%. Similarly, Zhang et al. [105] proposed a 10 kW adiabatic micro-CAES system for wind power peak shaving in residential applications, employing scroll compressors and expanders and achieving a daily energy efficiency of 47.35%.

Beyond the three classical CAES configurations, hybrid and integrated architectures have emerged as a significant research trend, particularly at small scale, with the aim of overcoming intrinsic efficiency limitations and increasing overall energy utilization. In hybrid CAES systems, compressed air storage is combined with additional energy conversion or storage technologies, enabling multi-energy outputs and improved operational flexibility [111].

One of the most investigated hybrid solutions, trigenerative CAES (T-CAES) systems aim to simultaneously produce electricity, useful heat, and, in some cases, cooling by recovering compression heat and exploiting the cooling effect associated with air expansion. Compression heat can be supplied to external thermal loads, while expansion-induced cooling may be used for refrigeration or air-conditioning [27, 112]. For example, Castellani et al. [113] designed a 1.5 kW micro-CAES system for combined cooling, heating, and power generation in historical buildings in Italy, achieving an overall energy efficiency of approximately 26%. Although several other trigenerative configurations have been investigated, reported electrical efficiencies at very small scale remain generally modest—often below 30%—with exergetic efficiencies typically in the range of 40–45% or

lower, depending on system layout and operating conditions [28, 29, 114].

Another relevant hybrid solution involves the integration of CAES with Organic Rankine Cycles (ORC) or Kalina cycles to convert compression heat or other low-grade thermal energy into additional electrical power, operating either in parallel or in cascaded configurations. Javidmehr et al. [33] proposed a highly integrated system combining CAES, a micro gas turbine, an ORC, a solar dish collector, and a multi-effect distillation unit, achieving an overall efficiency of up to 70% at a rated power of 100 kW. Similarly, Fu et al. [115] reported significant performance improvements for the TICC-500 pilot plant by adopting a series-parallel operating strategy for compressors and expanders coupled with an ORC, with the overall cycle efficiency reaching 70.53%.

CAES has also been proposed in hybrid configurations with conventional power generation units operating in combined heat and power (CHP) mode. In these architectures, CAES may function both as an energy storage system and as an auxiliary subsystem, for instance by supplying pressurized air or by recovering available thermal streams. These solutions are particularly attractive for CHP plants or for the retrofitting of off-grid systems powered by diesel generators, where CAES can contribute to fuel savings, emission reduction, and enhanced operational flexibility [31, 107, 116].

Despite their potential advantages, hybrid CAES architectures are often affected by increased system complexity, higher capital costs, and non-optimal component scaling at small size, which can negatively impact both efficiency and economic viability. Moreover, their effective operation requires advanced energy management and control strategies to handle strong coupling between subsystems and to avoid performance degradation under variable operating conditions [28, 45, 47, 117].

Experimental investigations on micro-scale CAES systems remain relatively scarce compared to simulation-based studies. Maia et al. [118] investigated an expansion unit based on a modified automotive turbocharger; however, the achieved energy conversion efficiency was limited to approximately 45%. More recently, Lu [119] reported experimental results for a

complete micro-CAES system, achieving an overall energy efficiency of 42.3%. Cheayb et al. [44] developed and experimentally validated a micro-CAES prototype with an output power between 1 and 2 kW, reporting overall efficiencies below 17%, primarily due to component inefficiencies and thermal losses.

Overall, the literature on micro and small-scale CAES depicts a fragmented landscape, in which system performance is strongly influenced by scale effects and architectural choices. Compared to large-scale systems, lower efficiencies are observed, with a clear degradation of performance as system size decreases. Experimental investigations remain limited and broadly confirm these trends, reporting efficiencies that are comparable to or lower than those predicted by theoretical and numerical studies. Hybrid and integrated architectures show the potential to enhance overall energy utilization; however, the associated increase in system complexity and sensitivity to operating conditions often offsets the expected performance gains.

### **2.3.2 Key components and technological challenges**

The design of micro and small-scale CAES systems differs substantially from that of large-scale installations, as it must reconcile compactness, flexibility, cost constraints, and acceptable efficiency levels [26]. Unlike utility-scale CAES plants, which benefit from underground air storage and purpose-designed turbomachinery, small-scale systems operate with limited storage volumes and experience higher off-design operation, making component selection and integration a critical challenge.

High-pressure vessels are the most commonly adopted air storage solution in small-scale CAES systems. Steel tanks, composite vessels, and repurposed industrial pressure equipment are typically used, with operating pressures ranging from several tens to several hundreds of bar. The isochoric nature of such storage results in large pressure variations during charging and discharging, forcing compressors and expanders to operate predominantly under off-design conditions. This behavior degrades efficiency and often requires throttling or pressure regulation devices,

introducing additional irreversibilities. Consequently, storage volume, pressure range, and discharge duration become key design parameters, markedly different from those of large-scale CAES systems [26, 59].

Compressors and expanders represent another major technological challenge. Positive displacement machines, including reciprocating, screw, and scroll devices, are most frequently employed due to their relatively good performance at low power ratings and partial-load operation [59, 112]. Turbomachinery-based solutions are less common at small scale, mainly due to cost and efficiency penalties at small scale. Moreover, many experimental small-scale CAES systems rely on commercially available components not specifically designed for CAES applications, resulting in suboptimal component efficiency [120]. Compressors and expanders specifically designed for small-scale CAES applications have only been investigated in recent years and have not yet reached an industrial level of development [121, 122].

Thermal management further constrains system performance. Heat exchangers are required to reject compression heat and reheat the air before expansion, making them critical components. Plate-fin and double-pipe heat exchangers are generally considered suitable for micro-CAES due to their compactness and heat transfer performance. Nevertheless, unresolved issues remain, particularly related to pressure losses in compact exchangers.[26].

Overall, the literature identifies system components as one of the main technological bottlenecks for small-scale CAES. The lack of components specifically designed for CAES operation leads to efficiency penalties, increased losses, and limited economic viability, highlighting the need for dedicated component development.

### 2.3.3 Applications of small-scale CAES systems

The application domain of small-CAES is shaped by its key characteristics, such as modularity, long service life, low self-discharge rate, and the ability to couple electrical and thermal energy domains. Unlike large-scale CAES plants designed for centralized grid support, small-scale systems

are mainly deployed in distributed, behind-the-meter, and multi-energy contexts [26, 35, 111]. In these applications, operational flexibility and multi-service capability are often more relevant than high round-trip efficiency, with systems typically operating under frequent start–stop conditions [123, 124]. In the following, an overview of the main application areas is provided, structured according to the field of use, however, that alternative classification criteria are possible (e.g., based on system functionality or service provision), and that many of the identified application scenarios are closely interconnected.

Integration into micro-grid constitute probably the most promising application area for small-scale CAES, both in grid-connected and islanded configurations. In grid-connected microgrids, CAES contributes to balancing supply and demand, mitigating renewable intermittency, and enhancing operational flexibility [27, 111]. In these contexts, CAES can support multiple value streams, including self-consumption maximization, load regulation and the provision of ancillary services ensuring the security and stability of micro grids [26]. In islanded or weakly interconnected microgrids, such as remote communities or telecommunication infrastructure, long service life and low maintenance requirements are critical advantages of CAES technology [107, 125]. To enhance the dynamic response of such system, CAES can be combined with fast-responding storage technologies, such as batteries or supercapacitors, are frequently proposed [27, 68].

Behind-the-meter applications represent another of the most investigated deployment scenarios for small-scale CAES. In these systems, CAES is typically integrated with on-site renewable energy generation, mainly photovoltaic systems, to increase self-consumption and reduce reliance on the public grid enabling time-shifted energy utilization [113, 124]. Although small-scale CAES generally exhibits lower round-trip efficiency compared to electrochemical storage technologies, its deployment can be economically and energetically justified when excess renewable electricity has low or negligible marginal cost [126]. In addition to electrical storage, behind-the-meter CAES systems can be embedded in cogeneration or trigeneration architectures for space heating, domestic hot water, or cool-

ing applications [127]. Nevertheless, practical challenges related to space requirements, noise mitigation, and system reliability remain relevant, particularly in densely populated environments [113] .

Small-scale CAES has also been proposed for industrial applications, particularly in facilities characterized by intermittent electricity demand and the availability of waste heat. In such environments, CAES can operate simultaneously as an electrical energy storage system and as a means of valorizing low to medium-temperature thermal energy by recovering and reusing compression heat during the expansion phase [128–130].

Other potential, though less extensively analyzed, application areas include participation in local energy markets, where small-scale CAES systems may enable the buying and selling of electricity and, where regulatory frameworks allow, thermal energy within energy communities or district-level networks [26]. The literature also addresses applications in the transport sector, encompassing both the use of compressed air for vehicle propulsion and the recovery of braking energy through compressed air-based regenerative systems. However, these applications remain relatively underdeveloped and regarded as only marginally promising [131, 132].

Overall, the analyzed application scenarios indicate that small-scale CAES systems are mainly deployed in distributed contexts characterized by a strong interaction with variable renewable energy sources and dynamic load profiles. As a result, their operation is often dominated by frequent cycling and off-design conditions rather than steady-state performance, condition that could negatively affect CAES efficiency, highlighting the importance of advanced control strategies and coordinated energy management [26]. These considerations suggest that the assessment of small-scale CAES systems requires modeling approaches capable of capturing dynamic behavior, off-design operation, and system-level interactions, aspects that are only partially addressed in the current literature as discussed in the following sections.

## 2.4 Modeling and Optimization Approaches for CAES Systems

The modelling literature on CAES system is characterized by a progressive shift from purely thermodynamic, cycle-level analyses toward integrated, multi-domain formulations that explicitly represent component behaviour, dynamic operation, and system-level supervisory control [27].

At the highest level of abstraction, several studies adopt simplified modelling choices such as constant efficiencies, lumped thermal capacities, and quasi-steady operation [45, 133, 134]. These models are useful for screening system architectures and for identifying first-order sensitivities with respect to storage pressure, staging, and thermal management. However, the underlying assumptions may obscure key off-design penalties when systems are operated under realistic duty cycles. Simplifications that may be acceptable for large-scale CAES installations can become misleading when extrapolated to micro-CAES applications. At small scale, in fact, the relative impact of non-idealities is significantly amplified, leading to performance deviations that cannot be neglected [26, 35, 40]. In response to these limitations, more recent works implement semi-empirical component maps or physics-informed performance correlations for compressors and expanders, allowing efficiencies to vary with pressure ratio, rotational speed, and mass flow rate [43, 110, 135]. Such formulations provide a more accurate representation of part-load and off-design regimes, which are ubiquitous in distributed energy applications. A recurring outcome of these studies is that constant-efficiency assumptions tend to overestimate system performance, particularly when the CAES system is dispatched with frequent power setpoint changes or shallow charge–discharge cycles. A further fundamental aspect concerns the representation of CAES operation under realistic operating conditions. This issue is especially relevant for micro and small-CAES systems deployed in behind-the-meter applications such as renewable generation smoothing, microgrid dispatch, and demand response, where the system rarely operates at nominal design conditions. Instead, it must continuously track time-varying power com-



mands dictated by fluctuating renewable generation, load profiles, or ancillary service requirements [26]. By contrast, a common practice in the literature is to characterize system behaviour through an idealized full charging phase, from minimum to maximum allowable storage pressure, followed by a full discharge cycle [7, 47, 136].

Beyond thermodynamic cycle analysis, a key aspect in CAES development is system sizing and optimization. This problem is typically formulated as a coupled task involving: (i) component sizing (compressor/-expander rated power or displacement, heat exchanger area, and thermal energy storage mass or volume), (ii) storage sizing (volume and maximum allowable pressure), and (iii) operating policy, including dispatch and control parameters. Within this framework, optimization objectives vary across the literature. Many works target round-trip efficiency or specific energy [48, 98, 137]. Others prioritize economic metrics, such as levelized cost of storage or net present value, leading to designs that may sacrifice peak efficiency in favour of economic aspect [110, 138]. Multi-objective formulations are increasingly used to expose trade-offs among efficiency, cost, environmental aspect, often revealing Pareto fronts [107, 139]. A consistent insight from these studies is that optimizing the thermodynamic cycle in isolation is insufficient, as optimal CAES design depends strongly on the intended use case and duty cycle—such as cycling frequency, depth of discharge, and peak power requirements. Consequently, the appropriate sizing of rated power and energy storage capacity has a decisive impact on system operation and profitability, requiring application-specific techno-economic and environmental optimization to avoid under- or over-sizing [47, 140]. Nevertheless, as noted by Bazdar et al. [27], only a limited number of studies have addressed the optimal sizing of CAES systems, either through analytical optimization of compressor/turbine size and air storage volume or by determining optimal power and energy capacities within integrated energy systems.

Methodologically, the literature spans parametric sweeps, gradient-free heuristics, and mixed approaches. Parametric studies remain common because they offer interpretability and can map sensitivities to key de-

sign variables such as pressure ratio, number of stages, and intercooler effectiveness [141]. However, for integrated systems with many coupled variables and constraints, heuristic or evolutionary optimization has been employed to explore large design spaces without strong convexity assumptions [28, 98]. Optimal scheduling of distribution power systems in the presence of CAES have been carried out mainly through the use of bi-level programming approach [142, 143], co-optimizing of design and dispatch. Advanced optimization approaches, such as genetic algorithms and bi-level programming, have been applied mainly to large-scale or small-network CAES systems (at least 2–5 MW). Across these studies, a recurring challenge is the trade-off between modelling fidelity and computational tractability: high-fidelity models capture part-load and dynamic losses but are often impractical for long-horizon optimization, whereas simplified models enable scenario analysis at the cost of under-representing inefficiencies [138, 143].

Overall, the modelling and optimization literature highlights that CAES system, and particularly small-CAES, cannot be robustly assessed with purely ideal thermodynamic calculations: realistic conclusions require explicit representation of off-design component performance, thermal subsystem dynamics, and constraint-aware control. At the same time, the field continues to face a methodological challenge in balancing fidelity and computational efficiency, particularly for co-optimization and long-horizon operational studies

## 2.5 Economic and Environmental Assessment of CAES Systems

The economic and environmental assessment of CAES systems represents an increasingly relevant research area. While thermodynamic feasibility and performance improvements dominate early-stage investigations, the transition toward comprehensive economic and environmental analyses is essential to properly evaluate the viability and sustainability of such systems.

### 2.5.0.1 Techno-economic assessment

Economic aspects of CAES systems are predominantly investigated through techno-economic analyses, with most studies focusing on large-scale, cavern-based installations. For such systems, reported levelized cost of storage values typically range between 50 and 150 \$/MWh, assuming long service lifetimes (often exceeding 30 years), high capacity factors, and favourable electricity price differentials[2, 144, 145]. However, these assumptions are rarely valid for small-scale CAES systems, whose economic performance is constrained by different technological, operational, and market conditions.

At small scale, CAES systems rely almost on aboveground pressurised vessels and modular mechanical components, resulting in substantially higher specific capital costs. Reported specific capital costs generally range between 200 and 300 \$/kWh of installed storage capacity, depending on pressure level, vessel material, and system layout[27]. Consequently, LCOS values for small-scale CAES are typically higher, often exceeding 150–400 \$/MWh, making economic viability strongly dependent on utilisation rate, cycling frequency, and system lifetime assumptions[110, 134, 145]. Capital expenditure dominates the cost structure of small-scale CAES systems, with pressure vessels and mechanical components representing the main investment drivers due to limited economies of scale [27, 124, 146]. However, efficiency losses significantly affect operating costs by increasing charging electricity demand, making economic performance strongly dependent on round-trip efficiency and electricity price dynamics or self-consumption schemes[110, 124, 147].

As discussed in the previous section, many studies still rely on simplified modelling approaches based on fixed nominal round-trip efficiencies, typically in the range of 50–70%, particularly when moving from purely thermodynamic analyses to economic assessments. Such assumptions neglect the fact that small-scale CAES systems often operate under partial-load conditions, leading to a systematic underestimation of performance losses [145, 147, 148]. As a consequence, indicators such as the levelized cost of storage and net present value may be overly optimistic

[27].

Several studies have also compared small-scale CAES with battery-based storage technologies. While lithium-ion batteries generally exhibit lower storage costs for short-duration applications, CAES may offer advantages for longer storage durations, as well as in terms of longer lifetime and reduced performance degradation [134, 145].

Overall, the literature indicates that the economic viability of small-scale CAES systems is highly sensitive to modelling assumptions regarding efficiency, capacity factor, electricity price dynamics, and component lifetime. The widespread adoption of static, nominal-point models limits the robustness and comparability of reported LCOS and NPV values[2, 108]. Incorporating dynamic, off-design performance, realistic cycling profiles into techno-economic assessments is therefore essential to provide credible economic indicators for micro and small-scale CAES applications.

#### 2.5.0.2 Environmental impact and life cycle assessment

From an environmental perspective, Life Cycle Assessment studies on compressed air energy storage systems, and on energy storage technologies more broadly, have increased in recent years. Nevertheless, they remain limited when compared to techno-economic literature. In the review conducted by Rahman et al. [50], only about one third of the analyzed energy storage studies adopt an LCA-based approach, with CAES representing only a minor subset of the assessed technologies. Reported results exhibit a wide variability in the environmental performance of CAES systems, with global warming potential (GWP) values ranging from approximately 20 to 740 gCO<sub>2</sub> eq./kWh. This variability is largely driven by differences in assumed electricity mixes, system efficiencies, and plant scale.

The majority of existing LCA studies focus on utility-scale CAES installations. For large adiabatic CAES systems, Kapila et al. [51] and Oliveira et al. [52] show that life-cycle impacts are predominantly governed by the operational phase when fossil-based electricity mixes are considered, whereas construction-related impacts become significant only under low-carbon charging scenarios. In contrast, LCA studies addressing small-

scale CAES systems remain scarce and methodologically heterogeneous. For instance, Cocco et al. [53] compared the environmental impacts associated with supplying the electricity demand of a small town (10 MW) through either a photovoltaic (PV) plant or a combined PV–CAES system, against a reference scenario in which the demand is fully met by the grid. Their results indicate that the lowest environmental impact is achieved by a standalone PV system (20 MW) without storage. Although the integration of CAES increases renewable self-consumption, it also leads to a higher overall environmental impact in the analyzed configurations. When CAES systems are downscaled, additional factors become critical from an environmental standpoint. Small-scale CAES systems typically rely on aboveground pressurized vessels, which are characterized by higher embodied emissions compared to geological storage solutions. Moreover, at sub-MW scale, CAES energy output is highly sensitive to part-load operation, which is typical of systems coupled with variable renewable energy sources [110]. As a consequence, the widespread use of nominal or design-point efficiency values in LCA studies risks underestimating the actual per-kWh environmental impacts of small-scale CAES systems. Furthermore, environmental performance indicators are often expressed in terms of avoided CO<sub>2</sub> emissions or embedded within broader techno-economic or system-integration studies, rather than being addressed through dedicated cradle-to-grave assessments [149, 150]. This practice limits the comparability and robustness of reported results, particularly for technologies operating under highly variable conditions.

Overall, while LCA is increasingly recognized as a necessary tool for assessing the sustainability of energy storage systems [151], robust and harmonized applications remain limited. Without explicitly accounting for dynamic operational behavior, comprehensive cradle-to-grave inventories, and appropriate functional units for multi-output systems, the current evidence base remains insufficient to fully demonstrate the environmental advantages of CAES in distributed energy storage applications [27, 110, 152].

## **2.6 Resume of the literature gaps and thesis positioning**

The review of the state of the art highlights that, despite the increasing interest in small-scale compressed air energy storage, the development of small and micro-scale CAES systems is still characterized by a fragmented and uneven knowledge base. Several gaps emerge across technological development, modelling practices, system integration, and sustainability assessment.

A first gap emerging from the literature concerns the scarcity of experimental results at small scale. Existing experimental activities are predominantly limited to proof-of-concept demonstrators or to the testing of individual components, often operated under controlled or nominal conditions. Only a limited number of studies report full-cycle experimental data under realistic duty cycles. This lack of experimental evidence introduces significant uncertainty regarding achievable efficiencies, operational reliability, and maintenance requirements, particularly in the presence of off-design operating conditions.

A further gap related to technological aspects concerns the limited availability of components specifically designed for small-scale CAES applications. Dedicated compressors and expanders for micro-CAES systems have only been investigated in recent years and have not yet reached an industrial level of maturity. As a consequence, experimental studies often rely on adapted or repurposed commercial machines, which are not optimized for CAES operation and therefore exhibit suboptimal performance. Conversely, numerical studies frequently assume machine efficiencies derived from large-scale or industrial applications, whose transferability to small-scale systems remains uncertain.

From a modelling standpoint, a further gap arises from the widespread adoption of simplified system modelling approaches in the CAES literature. Many studies rely on steady-state formulations and constant component efficiencies, despite small-scale CAES systems operating under strongly dynamic and off-design conditions driven by variable renewable generation

and dynamic load profiles. As a result, key performance indicators—such as round-trip efficiency, available power, and usable energy capacity—are highly sensitive to modelling assumptions. A related limitation concerns system sizing and optimization, which are often addressed in a fragmented manner, focusing on selected subsystems or assuming predefined operating strategies. Fully integrated frameworks that jointly optimize component sizing and operational control while accounting for off-design behaviour remain scarce. Although recent research has increasingly explored the integration of dynamic modelling and sizing optimization through advanced numerical approaches, such as genetic algorithms, neural networks, and bi-level programming, these developments have been predominantly applied to large-scale CAES systems. In contrast, studies adopting fully integrated, dynamic, and optimization-oriented frameworks for small-scale CAES, particularly at the sub-MW scale, remain limited.

These methodological limitations propagate into techno-economic and environmental assessments. The literature reveals a wide dispersion in reported economic indicators, such as specific investment costs or levelized cost of storage, which are strongly influenced by the assumed energetic performance of the system and, consequently, by the level of detail and accuracy adopted in its modelling. In particular, life cycle assessments of CAES systems remain scarce, moreover, environmental performance is frequently expressed in terms of avoided  $CO_2$  emissions computed on energy output of the storage, rather than being addressed through dedicated cradle-to-grave assessments. Fully integrated studies simultaneously accounting for thermodynamic performance, economic evaluation, environmental impacts, and system optimization therefore remain limited, especially for small-scale CAES applications.

Overall, the literature gap is therefore twofold. On the one hand, fundamental technological challenges—such as the lack of dedicated machinery, limited experimental validation, and the early maturity of small-scale CAES—remain largely unresolved and represent open research directions for future work. On the other hand, even within these constraints, there is a clear need for more consistent and realistic system-level assessment

methodologies, capable of accounting for off-design operation and the interaction between design and operation.

Within this context, the present thesis does not aim to address the technological gaps related to component development or large-scale experimental validation. Instead, it positions itself at the system modelling and assessment level, with the objective of providing a realistic and system-oriented evaluation of micro and small-scale CAES applications. The work focuses on architectures based on commercially available components and adopts a dynamic, off-design modelling approach to describe operation under realistic load and generation profiles. By combining dynamic simulations with integrated sizing, optimization, and techno-economic and environmental analyses, the thesis seeks to clarify what level of performance, efficiency, and sustainability can realistically be expected from technologically mature CAES systems when operated under real-world conditions.





## Chapter 3

# Numerical Modeling and Exergetic Analysis of a Micro-CAES Prototype

This chapter introduces the modelling and analysis of an industrial prototype of a micro-CAES system, which represents the starting point of this doctoral work. It explains the methodology used to develop and validate a dynamic model of the system in OpenModelica, taking advantage of the Modelica language for simulating time-dependent thermodynamic processes. The component models, governing equations, and main modelling assumptions are described. The model is then validated by comparing the simulation results with experimental data from the real system. An exergetic analysis of the validated model is also carried out to identify the main sources of irreversibility in the reference configuration. Overall, the chapter provides a clear modelling and diagnostic framework that supports the approach used in this thesis and highlights where improvements may be needed. This framework forms the basis for the system modifications explored in the following chapters.

### 3.1 System description

The baseline system for the present analysis is an adiabatic micro-CAES prototype, built and tested by GFM S.p.a [153]. A plant schematic representation is shown in Fig.3.1.

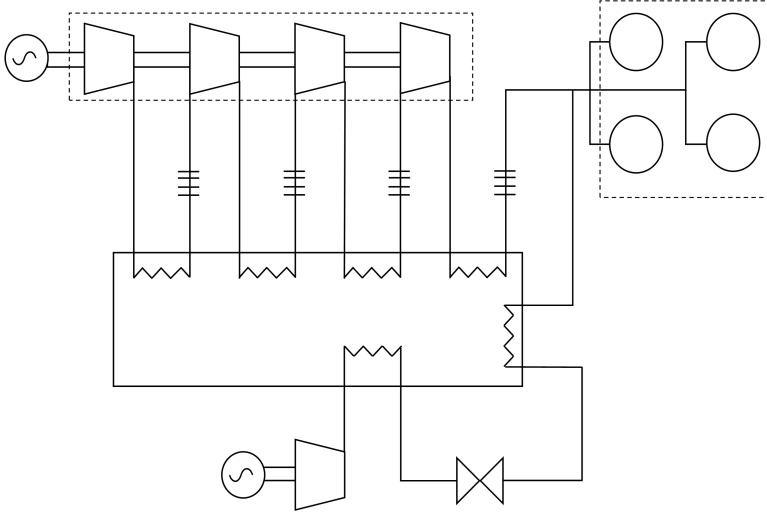


Figure 3.1: Scheme of the CAES plant

The system is mainly composed of a 4-stage reciprocating compressor, a thermal energy storage system, a series of high-pressure vessels, a lamination valve, and a micro-turbine. The compressor stages are air-cooled through ambient air moved by a fan connected to the electric motor that powers the compressor train, in order to limit the compressor parts operating temperature. The removed heat is dissipated to the ambient. Moreover, the compression is inter-cooled thanks to four heat exchangers immersed in an oil reservoir that works as a thermal energy storage and enables heat regeneration by supplying heat back to the air during the discharge phase. Additional heat exchangers (i.e. heat sinks), at the exit of the TES, allow the air temperature to further decrease, up to about

ambient temperature, before reaching the next compressor stage, through heat dissipation to the ambient.

Air is stored in a series of isochoric vessels, causing the compressor to operate under a variable pressure ratio, while the throttling valve regulates the turbine inlet pressure. The turbine delivers a rated output of 5 kW under nominal inlet conditions of 5 bar and 100 °C. The nominal storage pressure of 250 bar was selected to meet the target CAES capacity of 5 kW over a 12-hour discharge period.

## 3.2 Numerical modeling

In the development of this thesis work, the systems under investigation were modeled using the open-source software OpenModelica [154]. The following sections provide a brief overview of the main features of this software, highlighting the aspects that motivated its selection for this work.

### 3.2.1 Modelica Language

The Modelica language, first introduced in 1997 by Elmqvist and Mattsson [155], is an object-oriented, declarative, equation-based modeling language founded on differential-algebraic equations (DAEs). It provides a powerful and flexible framework for describing the dynamic behavior of complex physical systems across multiple engineering domains.

One of the most distinctive features of Modelica, setting it apart from most traditional engineering software, is its acausal modeling paradigm. The equations can be stated in a neutral form without consideration of computational order, i.e. non-causal modeling. In contrast, causal modeling approaches represent systems as interconnections of submodels defined in an explicit state-space or ordinary differential equation (ODE) form, where input–output causality is imposed a priori. General formulation of a causal problem is expressed by the following equations:

$$\frac{dx}{dt} = f(x, u) \quad (3.1)$$

$$y = f(x, u) \tag{3.2}$$

where  $u$  is input,  $y$  is output and  $x$  is the state. Normally, equations of the models need to be manipulated in order to get this form. In general, the equations describing a model must be manipulated and reformulated to obtain this explicit state-space form. Achieving such a formulation often requires significant analytical effort involving symbolic transformations and structural analysis. Differently, in Modelica language it is possible to write the equations in their natural form, i.e. as a system of differential-algebraic equations (DAEs):

$$F\left(x, \frac{dx}{dt}, y, u\right) = 0 \tag{3.3}$$

where  $x$  is the vector of unknowns that appear differentiated in the equation and  $y$  is the vector of unknowns that do not appear differentiated. Modelica environments automatically execute a series of symbolic and numerical algorithms behind the scenes, most notably index reduction, to transform the DAE system into a numerically solvable form. The resulting system is then integrated using suitable numerical solvers; by default, Modelica employs DASSL (Differential/Algebraic System Solver), an implicit, variable-order, multi-step method based on the backward differentiation formula.

Being an object-oriented language, Modelica enables the study of a system as a series of interacting components (objects). The overall system is decomposed into simpler, more manageable elements that are easier to analyze and understand. Each object encapsulates its own data, behavior, and structure, following the principles of encapsulation and modularity. Once the individual components have been defined, it is then necessary to specify the connections between them and, subsequently, their interactions and behavior. With these fundamental building blocks, complex models and hierarchical submodels can be constructed in an intuitive and systematic manner. In this modeling paradigm, models represent facts and relationships, rather than explicit data-processing

procedures. Consequently, in object-oriented modeling, models themselves are treated as objects, which can be reused, extended, and composed to represent increasingly complex systems. The interface of a Modelica model, called connectors, defines how the variables internal to the model interact with the external environment. Connecting multiple components through their connectors results in the creation of connection equations. These equations, depending on the specific Modelica keywords and connector types involved, can take different forms and convey different kinds of physical relationships.

As an illustrative example, the “Fluid connector” from the Modelica.Fluid library [156] is described below. This connector forms the fundamental interface used in the thermodynamic modeling of the developed models in the present work. This connector is characterized by the following physical quantities:

- Mass flow rate ( $\dot{m}$ )
- Pressure ( $p$ )
- Specific enthalpy ( $h$ )

When  $n$  Fluid connectors are connected, connection equations are generated to ensure the physical consistency of the interaction. Specifically:

- Conservation of Mass

$$\sum_{i=1}^n \dot{m}_i = 0 \quad (3.4)$$

- Equality of pressure

$$p_1 = p_2 = \dots = p_n \quad (3.5)$$

- Energy balance

$$\sum_{i=1}^n \dot{m}_i \cdot h_i = 0 \quad (3.6)$$

Given the characteristics described above, it is evident that Modelica provides a powerful and versatile framework for representing CAES systems, where simulating the time evolution of the thermodynamic state of the working fluid during the charging and discharging phases is of primary importance. Moreover, Modelica's architecture enables a high degree of model reusability across different system configurations and also facilitates the integration of thermodynamic models with other system aspects, such as control systems, mechanical and thermal aspects.

### 3.3 Thermodynamic model

This section presents the governing equations of the system model. The model is formulated as a lumped-parameter dynamic system and is solved in the time domain; however, most plant components are described through quasi-steady-state relations, since resolving the complete fast dynamics of the subsystems is beyond the scope of the present analysis. Such an approach is hereafter referred to as dynamic or time-varying modeling.

#### 3.3.1 Air Model

The gas model adopted to represent air depends on the specific component under analysis. For the compressor, turbine, pressure vessel, and heat exchangers, air is modeled as an ideal gas, whereas a real-gas formulation is employed for the throttling valve. The representation of air as an ideal gas in the analysis of CAES systems is widely adopted and generally accepted in the literature [32, 135]. In the present work, this assumption was primarily retained in order to reduce the computational effort required by the developed model. In particular, Calise et al. [135] performed a zero-dimensional modeling of a CAES system in TRNSYS and compared ideal and real-gas formulations. Their results showed that the ideal-gas approach led to an error of approximately 4% in the estimation of the compressor-train mechanical work, while reducing the computational time by about 56% with respect to the corresponding real-gas implementation.

A second aspect deserving discussion concerns the treatment of air humidity. In the present work, air is modeled as dry rather than humid, consistently with the assumptions commonly adopted in the modeling of CAES systems. Furthermore, in CAES plants, condensate is progressively formed and removed within the intercoolers of the multi-stage compression train. As a consequence, the air entering the storage vessel is already significantly dehumidified [60]. Nevertheless, residual moisture may still represent a critical issue during the expansion phase. In particular, when the air temperature drops below 0 °C, ice formation may occur within the expansion line, potentially causing operational issues and damage to the turbine. For this reason, in the studied system of the plant objectives is to maintain the expansion outlet temperature above the freezing point through appropriate thermal management strategies. However, as will be shown in the results section, the adopted configuration is not always able to satisfy this requirement, thereby motivating the introduction of further design modifications aimed at mitigating icing-related problems.

The ideal-gas model implemented in this work, available in the native Modelica library as `Modelica.Media.Air.DryAirNasa`, is based on the thermodynamic correlations reported by Gordon and McBride [157]. The model provides a temperature-dependent formulation of the specific heat capacity,  $c_p = f(T)$ , enabling an accurate evaluation of the thermodynamic properties of air over a wide temperature range.

For components in which the real-gas behavior of air plays a significant role, a real-gas formulation based on experimental thermodynamic data is adopted. This is the case of the throttling valve, where the pressure drop induces a non-negligible temperature variation governed by the Joule–Thomson effect, which cannot be accurately captured through an ideal-gas formulation. The adopted real-gas model, `Modelica.Media.Air.ReferenceAir.Air_ph`, relies on the thermodynamic data reported by Lemmon et al. [158]. The model expresses the thermodynamic properties as functions of pressure and specific enthalpy,  $(p, h)$ . Nevertheless, owing to the acausal nature of the Modelica language, the thermodynamic state can equivalently be determined by specifying alter-



native pairs of independent variables, such as temperature and pressure,  $(T, p)$ .

When two components characterized by different gas models are connected, the thermodynamic quantities affected by the adopted formulation, such as the specific enthalpy, are consistently recalculated at the interface by enforcing the continuity of pressure and temperature.

### 3.3.2 Air storage

The air is stored in a series of constant-volume pressure vessels. As air mass is progressively stored within the vessels, the storage pressure increases, leading to a corresponding increase in the pressure ratio required by the compressor. This mechanism inherently introduces a time-dependent behavior of the system. An analogous but opposite behavior occurs during the discharge phase, as the stored air mass decreases and the storage pressure progressively drops.

The pressure vessels are modeled by applying an internal energy balance to the gas stored within the control volume, as expressed in Equation (3.7). Given the large number of aboveground pressure vessels and the resulting significant heat losses, the air temperature inside the storage reservoir is assumed to coincide with the ambient temperature, following the approach adopted by Salvini et al. [147].

$$\frac{dU}{dt} = \dot{m}_{in} \cdot h_{in} - \dot{m}_{out} \cdot h_{out} + \dot{Q}_{amb} \quad (3.7)$$

### 3.3.3 Compressor

As introduced earlier, the compressor installed in the system is an air-cooled reciprocating machine. The thermodynamic process inside the compression chamber is inherently polytropic; however, in the present model it is approximated as an adiabatic compression followed by heat exchange with the surrounding environment. To analyse the compressor performance under different operating conditions, each compression stage is modelled by means of characteristic curves of volumetric and isentropic

efficiency, both expressed as a function of the pressure ratio. (dependency on rotational speed is not accounted for since the compressor is operated at fixed speed)

The mass flow rate processed by the compressor is given by:

$$\dot{m} = \rho_{in} \cdot V_s \cdot \eta_{vol} \cdot \frac{N}{60} \quad (3.8)$$

where  $\rho_{in}$  is the inlet density,  $V_s$  is the swept volume ( $\text{m}^3/\text{rev}$ ),  $N$  is the rotational speed in rpm, and  $\eta_{vol}$  is the volumetric efficiency accounting for clearance volume and leakage losses.

The power required by each compression stage is expressed as:

$$P_i = \dot{m} \cdot w_{is,i} \cdot \frac{1}{\eta_{is,i}} \quad (3.9)$$

where  $w_{is,i}$  denotes the isentropic specific work for the  $i$ -th stage (as defined in Equation (3.10)), and  $\eta_{is,i}$  is the corresponding stage isentropic efficiency.

The isentropic specific work required at each stage is evaluated assuming an adiabatic transformation:

$$w_{is,i} = \frac{\gamma}{\gamma - 1} \cdot R \cdot T_{in,i} \cdot \left( \beta_i^{\frac{\gamma}{\gamma-1}} - 1 \right) \quad (3.10)$$

where  $\beta_i = p_{out,i}/p_{in,i}$  is the pressure ratio of the  $i$ -th compression stage,  $\gamma = c_p/c_v$  is the ratio of specific heats,  $R$  is the gas constant, and  $T_{in,i}$  is the inlet air temperature for that stage.

The discharge temperature of the adiabatic compression is:

$$T_{ad,i} = T_{in,i} \cdot \left( 1 + \frac{\beta_i^{\frac{\gamma}{\gamma-1}} - 1}{\eta_{is,i}} \right) \quad (3.11)$$

while the actual discharge temperature, as a result of the compressor cooling, is estimated as:

$$T_{out,i} = T_{ad,i} - \frac{\dot{Q}_i}{c_p \cdot \dot{m}} \quad (3.12)$$

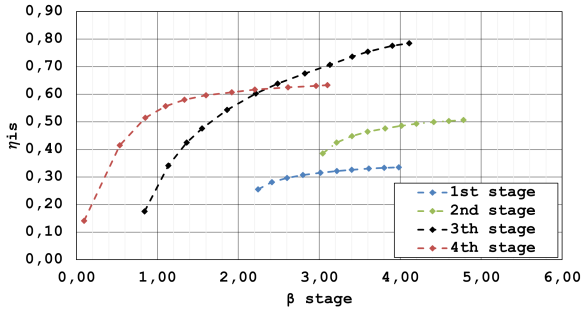


Figure 3.2: Compressor efficiency

where  $\dot{Q}$  is the instant heat flux released from the compressed air to the cooling air. This heat flux is computed by modelling an equivalent heat exchanger, where the two fluid streams are the cooling air and the compressed air at the compressor outlet. The overall heat transfer coefficient of this equivalent exchanger is estimated from experimental data and assumed to be constant throughout the simulations. This assumption is considered acceptable, since the mass flow rate remains nearly constant during operation. Both the volumetric and isentropic efficiencies were derived from experimental data collected during the operation of the CAES system. In particular, the isentropic efficiencies were derived in order to match the instantaneous compressor power and the measured outlet temperatures of each compression stage.

Fig.3.2 shows the isentropic efficiency in terms of pressure ratio for each compressor stage, as an example of characteristic curves used as input for the model.

### 3.3.4 Thermal storage

The thermal energy storage system consists of an insulated tank filled with mineral oil. As a first approximation, the temperature inside the

TES is assumed to be uniform and it is computed by an energy balance taking into account the heat exchanged with the compressed air and with the external environment. Equation (3.13) describes the energy balance, where  $\rho$  and  $c_{oil}$  are the oil density and specific heat capacity, respectively, assumed constant with temperature; their values are reported in Table 3.1.  $\dot{Q}_{hex}$  represents the total heat transfer rate exchanged with the air in the heat exchangers, while  $\dot{Q}_{amb}$  accounts for the heat exchanged with the external environment, computed according to Equation (3.14). The value of  $UA_{TES}$  is estimated from available experimental data and is considered to be constant with temperature.

$$\rho \cdot c_p \cdot \frac{dT}{dt} = \dot{Q}_{hex} + \dot{Q}_{amb} \quad (3.13)$$

$$\dot{Q}_{amb} = UA_{TES} \cdot (T_{amb} - T_{oil}) \quad (3.14)$$

|                              | Value |
|------------------------------|-------|
| Specific heat (J/kg)         | 1800  |
| Density (kg/m <sup>3</sup> ) | 817   |

Table 3.1: Mineral oil characteristic

### 3.3.5 Heat exchanger

The heat exchangers immersed in the thermal storage system essentially consist of shell-and-tube bundles. To model the heat exchanger (HEX), the  $\varepsilon$ -NTU approach is employed. This method, widely accepted and used in the literature [159], is based on the calculation of the exchanged heat rate ( $\dot{Q}$ ) as a fraction of the maximum possible heat rate ( $\dot{Q}_{max}$ ), as shown in Equation (3.15). The term  $\dot{Q}_{max}$  represents the maximum exchangeable heat, computed according to Equation (3.16), where  $C_{min}$  is the smaller of the two heat capacity rates of the fluids within the exchanger, and  $T_{in,h}$  and  $T_{in,c}$  denote the inlet temperatures of the hot and cold fluids, respectively. The heat exchanger effectiveness,

$\varepsilon$ , is a function of the number of transfer units (NTU), the heat capacity rate ratio, and the flow arrangement within the heat exchanger. Equation (3.17) provides the used definition of effectiveness, since the thermal capacity of the oil contained in the storage tank is much larger than that of the air flowing through the heat exchanger, it is assumed that  $\frac{C_{min}}{C_{max}} = 0$ . The NTU is defined as shown in Equation (3.18), where  $UA$  represents the overall heat transfer coefficient of the heat exchanger. The  $UA$  value of each heat exchanger is estimated from the available experimental data on the basis of the measured air temperatures at the inlet and outlet of the heat exchanger and of the TES temperature: the instantaneous  $UA$  value is first computed, and an average value is then adopted in the simulation. The resulting values are reported in Table 3.2, where the heat exchangers are numbered progressively from 1 to 4 for the compressor intercoolers, while indices 5 and 6 refer to the reheater upstream of the throttling valve and to the one upstream of the turbine, respectively. It is assumed that there is no pressure loss due to the heat exchangers, as the experimental data showed that these losses are negligible.

$$\dot{Q} = \dot{Q}_{max} \cdot \varepsilon \quad (3.15)$$

$$\dot{Q}_{max} = C_{min} \cdot (T_{in,H} - T_{in,C}) \quad (3.16)$$

$$\varepsilon = 1 - e^{-NTU} \quad (3.17)$$

$$NTU = \frac{UA}{C_{min}} \quad (3.18)$$

| UA<br>(W/K) |    |       |     |
|-------------|----|-------|-----|
| HEX-1       | 30 | HEX-4 | 10  |
| HEX-2       | 25 | HEX-5 | 110 |
| HEX-3       | 12 | HEX-6 | 110 |

Table 3.2: Heat exchangers assigned global heat transfer coefficient

### 3.3.6 Regulating Valve

The process of lamination in the regulating valve is assumed to be isenthalpic, as per Equation (3.19); therefore the enthalpy is kept constant between valve input and output. The use of a real gas model, as stated in Section 3.3.1, allow to evaluate the temperature decrease due to Joule-Thomson effect.

$$h_{in} = h_{out} \quad (3.19)$$

### 3.3.7 Micro-turbine

The turbine considered is a single-stage axial impulse turbo-expander. It was the subject of a previous experimental campaign, described in detail in [160], which provided an accurate characterisation of its operating performance. The specific work produced by the turbine is computed using the adiabatic expansion formulation (Equation (3.20)), as the experimental measurements demonstrated that the outlet temperature closely matches the adiabatic prediction.

The operating behaviour of the turbine is modelled through its characteristic curves of reduced mass flow (Equation (3.21)) and isentropic efficiency, both expressed as functions of the pressure ratio and rotational speed. Figure 3.3 reports the isentropic efficiency curves at three different inlet temperatures, illustrating the dependence of turbine performance on operating conditions. These correlations allow the determination of the operating point at each simulation time step, enabling an accurate prediction of mass flow, expansion work, and turbine efficiency.

$$w_{out} = \eta_{is}(\beta, rpm) \cdot \frac{\gamma}{\gamma - 1} \cdot R \cdot T_{in} \cdot \left( \frac{1}{\beta^{\frac{\gamma}{\gamma-1}}} - 1 \right) \quad (3.20)$$

$$M_r(\beta, rpm) = \frac{\dot{m} \sqrt{T_{in}}}{p_{in}} \quad (3.21)$$

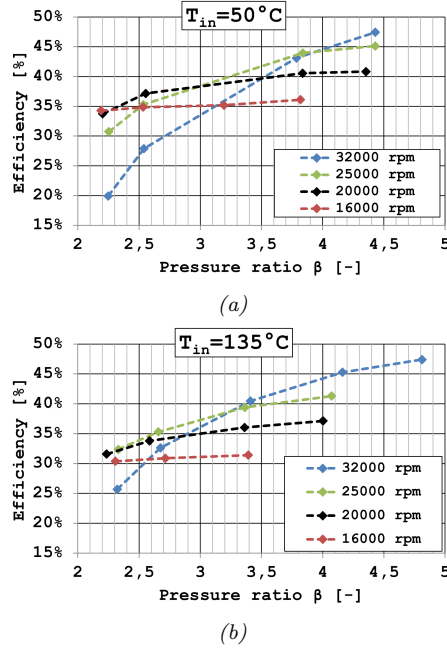


Figure 3.3: Micro turbine efficiency as a function of pressure ratio and rotational speed at different inlet temperature: (a) 50°C, (b) 135°C [160]

### 3.4 Energy and Exergy Analysis

An energetic and exergetic analysis of the system is carried out to evaluate the efficiency of the system and to identify the different sources of exergy loss and the impact of each of them.

Energy performance is assessed by the Round-Trip Efficiency (RTE), commonly used as one of the most important parameter to evaluate an energy storage system, and defined as per Equation (3.22), where  $E_{out}$  represents the energy supplied by the storage in the discharge phase while  $E_{in}$  is the total energy absorbed by the storage during the charge phase.

$$RTE = \frac{E_{out}}{E_{in}} \quad (3.22)$$

In addition to the energy assessment, an exergy analysis is also carried out. The latter is necessary because the former does not quantify the influence on the final efficiency of processes that do not involve an explicit energy transfer (e.g. throttling valve), nor the weight of heat fluxes and losses according to their temperature level.

In the present analysis, only the physical exergy is considered, while the contributions from chemical, kinetic, and potential exergy are neglected. The specific exergy of a flow stream (expressed in J/kg) is calculated according to Equation (3.23), whereas for components involving mass storage, such as the Thermal Energy Storage and the Compressed Air Storage (CAS), the system exergy is defined as shown in Equation (3.24).

$$ex = (h - h_0) - T_0(s - s_0) \quad (3.23)$$

$$ex_{st} = (u - u_0) + p(v - v_0) - T_0(s - s_0) \quad (3.24)$$

where the subscript 0 stands for reference condition (P=101325Pa and T=298.15K) For each component an exergy balance is imposed as reported in Equations (3.25) and (3.26) respectively for component without and with mass storage.

$$\sum \dot{Ex}_{in} - \sum \dot{Ex}_{out} - \dot{Ex}_D - \dot{Ex}_L - \dot{Ex}_W = 0 \quad (3.25)$$

$$\frac{dEx_{st}}{dt} = \sum \dot{Ex}_{in} - \sum \dot{Ex}_{out} - \dot{Ex}_D - \dot{Ex}_L - \dot{Ex}_W \quad (3.26)$$

where  $\dot{Ex}_w$  is the exergy flux associated with work,  $\dot{Ex}_D$  represents the exergy destruction rate, while  $\dot{Ex}_L$  denotes energy loss rate, in the form of heat transferred to the environment, calculated according to Equation (3.27):  $T$  is the temperature at which heat exchange occurs and  $\dot{Q}$  is the thermal power exchanged, considered positive when flowing from the external environment to the component.

$$\dot{Ex}_L = -\left(1 - \frac{T_0}{T}\right) \cdot \dot{Q} \quad (3.27)$$



The equations describing the exergy destruction rates for each component are reported in Table 3.3.

*Table 3.3: Exergy destruction and loss*

| Component        | Exergy destruction                                                                         |
|------------------|--------------------------------------------------------------------------------------------|
| Compressor       | $\dot{E}x_D = \dot{W}_{in} + \dot{E}x_{in} - \dot{E}x_{out} - \dot{E}x_L$                  |
| Expander         | $\dot{E}x_D = \dot{E}x_{in} - \dot{E}x_{out} - \dot{W}_{out}$                              |
| Heat exchanger   | $\dot{E}x_D = (\dot{E}x_{in,h} - \dot{E}x_{out,h}) + (\dot{E}x_{in,c} - \dot{E}x_{out,c})$ |
| CAS / TES        | $\dot{E}x_D = \dot{E}x_{in} - \dot{E}x_{out} - \dot{E}x_{ST} - \dot{E}x_L$                 |
| Regulating Valve | $\dot{E}x_D = \dot{E}x_{in} - \dot{E}x_{out}$                                              |

In the compressor exergy balance, both exergy destruction and exergy loss terms are included, since, as discussed in the previous section, the model accounts for heat dissipation to the ambient environment. For the turbine, only exergy destruction associated with the isentropic efficiency is considered, as the expansion process is assumed to be adiabatic. The temperature used to evaluate the compressor exergy losses corresponds to the thermodynamic mean temperature of the working fluid, calculated between the outlet temperature under ideal adiabatic compression and the actual outlet temperature, as defined in Equation (3.28).

$$T_{TA} = \frac{h_{ad} - h_{out}}{s_{ad} - s_{out}} \quad (3.28)$$

Finally, it's worthy to underline that an exergetic efficiency value for the entire system is not here introduced, since the system's only inputs and outputs are mechanical work; hence, its exergetic efficiency coincides with the round-trip efficiency previously defined.

### 3.5 Simulation performed

The developed model is employed for two types of analyses:

- In the first stage, aimed at model validation, the model is used to replicate the operating conditions of the experimental tests for which measurements were provided by the company. Both the charging and discharging phases are reproduced, as detailed below.
  - The simulated charging phase represents a complete charge, starting from an empty air reservoir and a TES at ambient temperature, and continuing until the rated pressure of 250 bar is reached within the storage vessels. During the charging four sets of storage vessels are filled sequentially: each set begins to be charged only after the previous one had reached its maximum pressure.
  - For the discharging phase, in order to reproduce the available experimental test, the simulation is initialised at the nominal full-storage pressure with a TES temperature of 140 °C; the latter being achieved during the experiment by means of an electric immersion heater. Consistently with the experimental campaign, the simulation is carried out over a duration of 5 h 15 min. Throughout the discharge, the throttling valve is regulated so as to keep the output power as close as possible to the rated value of 5 kW. Conversely to the charging test, the four sets of vessels are emptied in parallel.
- In the second stage, once the model has been validated, it is applied to assess the system performance over a complete operating cycle, consisting of a full charge followed by a full discharge (i.e., from empty to nominal storage pressure, and subsequently from full back to empty). In this case, the discharging phase is initialised using the boundary conditions (the pressure and temperature of the air reservoir and the TES temperature) obtained at the end of the charging simulation. Differently from the previous case, the

throttling valve is set to maintain the turbine inlet pressure at 5 bar. Once the storage pressure drops below this value, the expansion continues without throttling until the pressure within the storage tank reaches 2 bar, below which turbine operation is no longer recommended.

Table 3.4 summarises the simulations carried out and the corresponding boundary conditions employed in both the validation and exergetic analyses. As shown in the table, the compressor is operated at the fixed rotational speed used during the experimental campaign. In contrast, the turbine is run at a constant speed during the validation process, whereas in the exergetic analysis it is regulated according to the control strategy developed during the characterisation campaign conducted at our university [160]. This strategy determines the optimal rotational speed that maximises efficiency as a function of the pressure ratio and corresponds to the envelope of the performance curves reported in Figure 3.3.

Table 3.4: Simulation set-up

|                  | Charge phase         |                     |                     | Discharge phase      |                     |                  |                  |                        |
|------------------|----------------------|---------------------|---------------------|----------------------|---------------------|------------------|------------------|------------------------|
|                  | $p_{start}$<br>[bar] | $p_{stop}$<br>[bar] | Compressor<br>[rpm] | $p_{start}$<br>[bar] | $p_{stop}$<br>[bar] | Stop time<br>[h] | Turbine<br>[rpm] | $T_{oil}(t=0)$<br>[°C] |
| Model validation | 1                    | 250                 | 1780                | 250                  | –                   | 5h 15'           | 32000            | 140                    |
| Exergy analysis  | 1                    | 250                 | 1780                | 250                  | 2                   | –                | optimized        | 68                     |

It is worth to briefly recall that, due to the acausal nature of the Modelica language, no explicit sequence for solving the resulting system of equations is prescribed. Instead, the simulation environment automatically transforms the system of differential algebraic equations into a set of executable instructions behind the scenes. For instance, during the compression phase, the interconnection of the compression stages determines the pressure in the storage tank, which is given by:

$$p_{storage} = p_{atm} \cdot \beta_1 \cdot \beta_2 \cdot \beta_3 \cdot \beta_4 \quad (3.29)$$

Moreover, the connector connection enforces mass-flow continuity across all connected components. These aspects, combined with the look-up tables containing the characteristic curves of each stage, allow the software to determine the operating point of each compressor section. In practice, at every simulation time step, a pressure ratio and an efficiency value are assigned to each stage. This, in turn, enables the calculation of the absorbed power, discharge temperature, and dissipated heat.

## 3.6 Results and discussion

### 3.6.1 Model validation

The comparison between the measurements collected during the test campaign and the results produced by the mathematical model is reported in this section. Time-resolved measurements of mass flow rate, pressure, and temperature served as the main benchmarks. Figure 3.4 shows the locations of the instrumentation that provided usable measurements during the test. The data acquired from these instruments on the industrial prototype were supplied by the industrial partner; however, no detailed information on the measurement chain or on the associated uncertainty was made available. For this reason, the experimental results are assumed to be the reference, and the deviation of the simulation results from these data is assessed accordingly. In the following, a graphical comparison is presented between the temporal evolution of the quantities recorded by selected instruments during the experimental campaign and the corresponding values obtained from the simulation. The data are reported together with the percentage deviation of the simulation results from the experimental measurements at each sampling time.

Regarding the compression phase, Figure 3.5 and Figure 3.6 report the compressor pressure ratio and the discharge temperature at each compression stage, respectively. The experimental pressure ratio is computed as the ratio between the pressure measured at the inlet of a given compressor stage and that measured at the inlet of the preceding stage, thereby neglecting the pressure drop across the intercoolers and heat sinks.

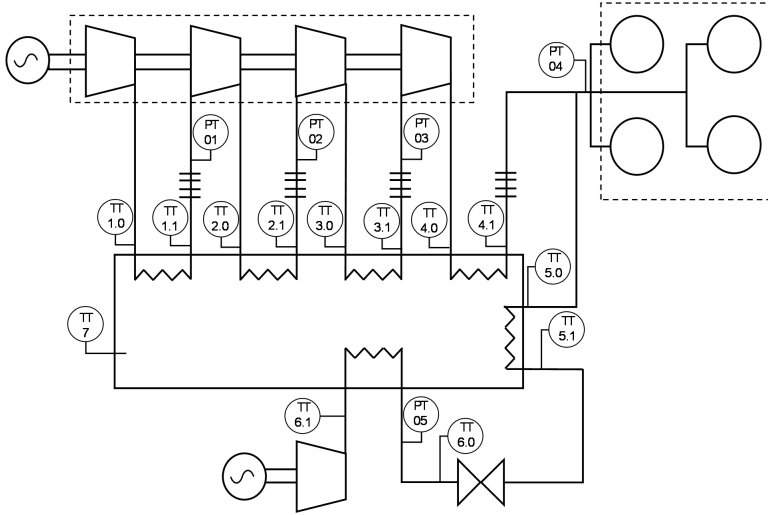


Figure 3.4: Simplified piping and instrument diagram of the industrial prototype

The comparison between experimental and numerical results shows good agreement. In particular, the relative error on the pressure ratio remains limited, in the range of 3–5%, whereas the deviation in terms of discharge temperature is somewhat larger (5–15%). This is attributable to the modelling approach adopted for the compression process (adiabatic compression followed by heat release), and remains within an acceptable range for the purposes of the present analysis. From a more global perspective, Figure 3.7 and Figure 3.8 show the pressure at the inlet of the air storage and the compressor power, respectively. For both quantities the relative error is very limited; this is a particularly relevant outcome, as these two parameters are the most significant in assessing the charging time and the round-trip efficiency of the system. Finally, Figure 3.9 reports the TES temperature, which exhibits good agreement with the experimental data, with a relative error below 10% for most of the simulated period and approaching negligible values towards the end of the simulation. The

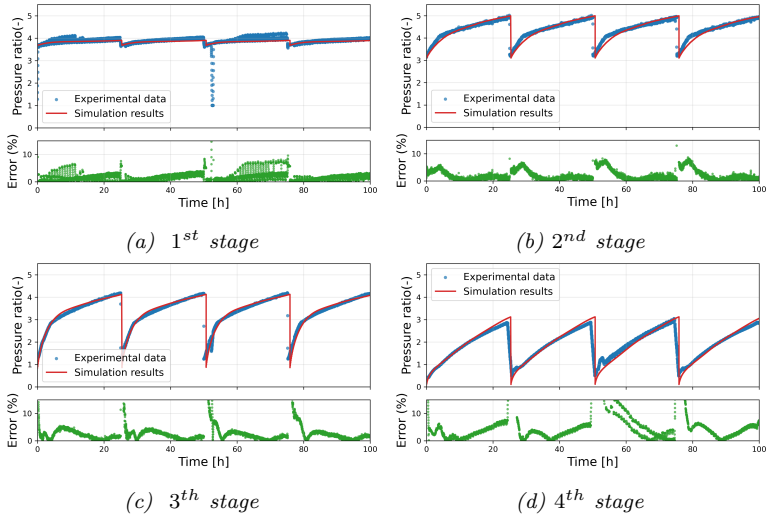


Figure 3.5: Compressor stages pressure ratio

amount of stored thermal energy is therefore estimated with satisfactory accuracy.

Regarding the discharging phase, a comparable level of accuracy is achieved. Figure 3.10 reports the turbine output power, which, in accordance with the regulation strategy adopted, remains nearly constant at approximately 5 kW. Figure 3.11 shows the turbine inlet pressure, which exhibits a slight increase throughout the discharge: as the inlet air temperature progressively decreases, a higher inlet pressure is required to maintain a constant output power. In general, the experimental data available for the discharging phase appear more dispersed and are affected by larger deviations than those of the charging phase. This behaviour is attributable to a non-ideal regulation of the valve, whose output pressure oscillates constantly; these fluctuations are directly transmitted to the turbine output power, which exhibits a comparable scatter. Nevertheless, the discrepancy between measured and simulated data remains within approximately 10%. Finally, Figure 3.12 shows the comparison for the

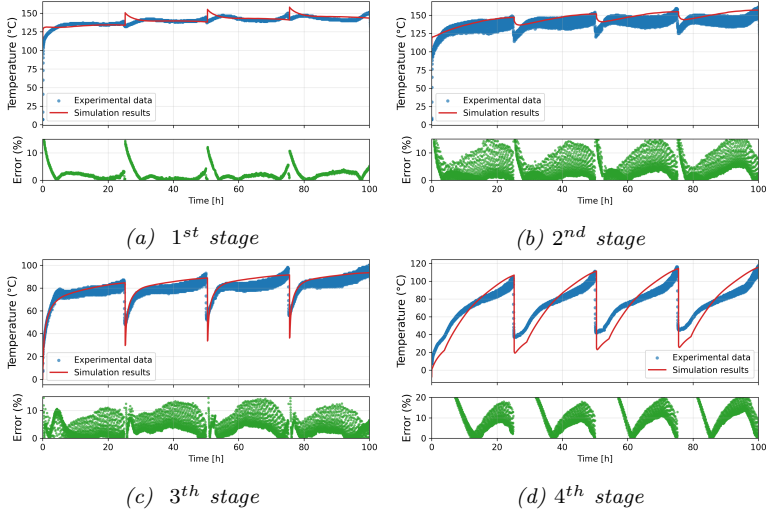


Figure 3.6: Discharge temperature of compressor stages

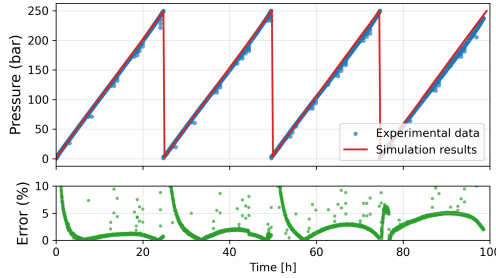


Figure 3.7: Compressor discharge (i.e. storage) pressure

pressure evolution at the outlet of the storage vessels during the discharge process, displaying acceptable agreement within the same error range.

Overall, the comparison indicates that the model accurately reproduces the time-dependent behaviour of the physical system, with only limited deviations from the experimental evidence. While this validation cannot be regarded as exhaustive nor extendable to conditions beyond those

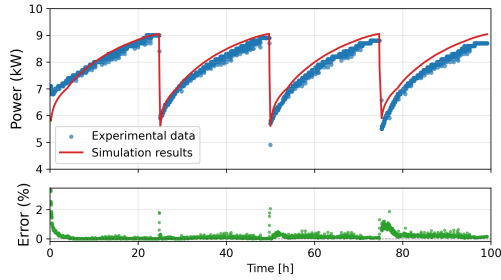


Figure 3.8: Compressor absorbed power

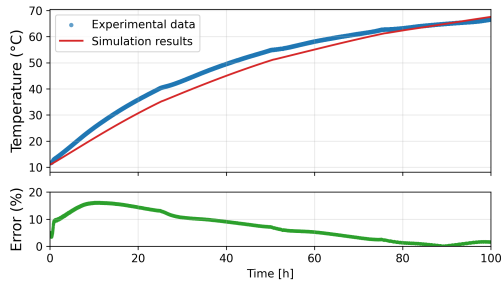


Figure 3.9: TES Temperature

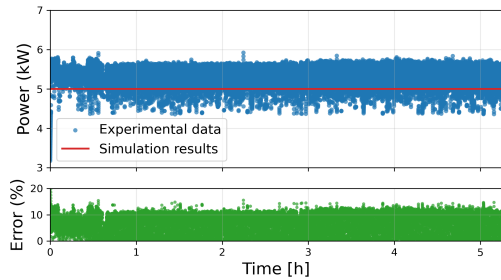


Figure 3.10: Turbine output power

investigated here, it demonstrates that the Modelica-based framework captures the temporal evolution of the relevant physical quantities with



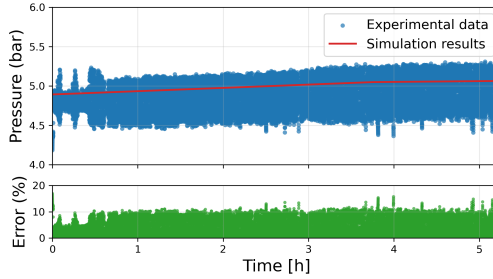


Figure 3.11: Turbine inlet pressure

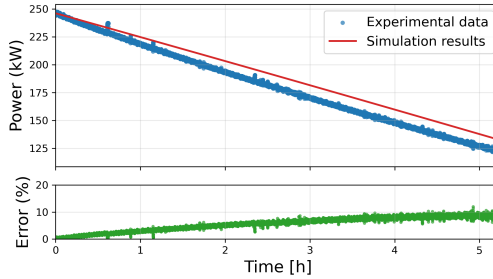


Figure 3.12: Storage pressure during discharge phase

satisfactory accuracy. Moreover, the modelling approach relies on well-established formulations, such as adiabatic compression and the  $\varepsilon - NTU$  method, which do not require extensive independent validation. The present comparison is therefore intended to confirm the correct implementation and integration of the sub-models within the overall system simulation, rather than to validate the underlying physical assumptions.

### 3.6.2 Energy and Exergy analysis

The results of the exergetic analysis are summarised in the pie chart shown in Figure 3.13 and in the Sankey diagram presented in Figure 3.14, where all energy and exergy flows are expressed as a percentage of the total absorbed energy. The pie chart highlights the remarkably low round-trip

efficiency of the system, approximately 6%, as the useful mechanical work recovered during the discharge phase directly corresponds to the Round-Trip Efficiency. The graph also reveals the three dominant sources of losses: exergy destruction due to compressor inefficiencies, heat dissipation from the compressor to the ambient environment, and exergy destruction across the throttling valve. These three mechanisms account respectively for 29%, 28%, and 25% of the total absorbed energy. A significant portion of the exergy is therefore lost as heat rejected by the compressor. Since adiabatic compression cannot be achieved in practice (cooling is required to ensure safe and reliable operation) this thermal energy cannot be stored in the TES and is instead dissipated to the surroundings. The huge amount of throttling losses arise from the system layout, in which the air is stored at up to 250 bar but expanded only from 5 bar at the turbine inlet. Consequently, reducing the pressure drop across the throttling valve, by lowering the storage pressure and/or increasing the turbine inlet pressure, appears essential to improve the overall energy efficiency of the system.

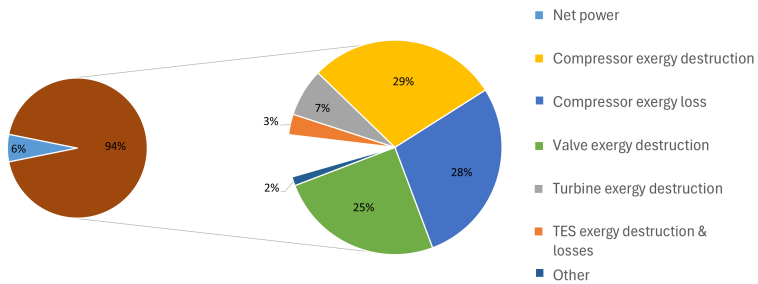


Figure 3.13: Exergy destruction and losses distribution

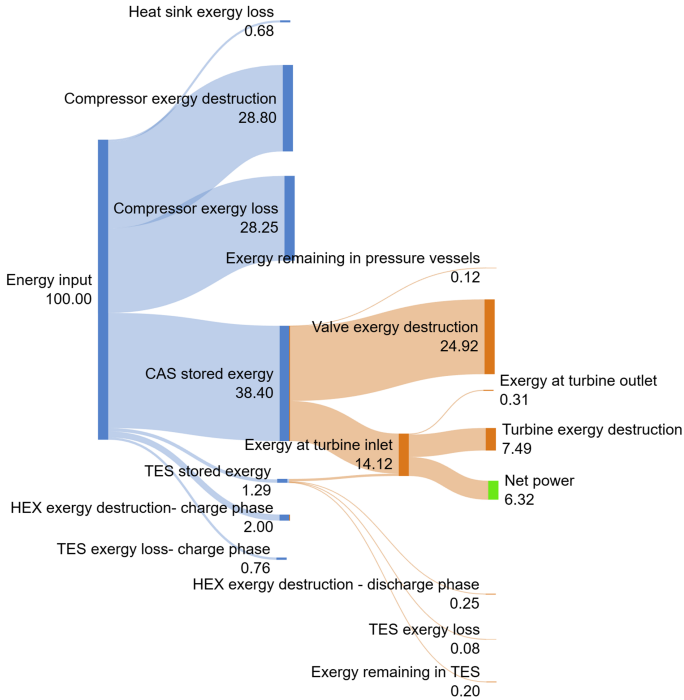


Figure 3.14: Sankey diagram describing exergy flows

A more detailed understanding of the exergy flows can be obtained from the Sankey diagram. This representation makes it possible to trace the exergy flows across the individual components and during both the charging and discharging phases. For example, the turbine accounts for approximately 7.5% of the total absorbed energy in the form of exergy destruction. This value corresponds to nearly 50% of the exergy entering the turbine, clearly indicating its limited efficiency. The Sankey diagram also shows that the amount of exergy stored in the TES is negligible when compared to that stored in the pressure vessels. While only 1.29% of the

absorbed energy is stored in the TES, the pressure vessels retain 38.40%. Combined with the large fraction of heat dissipated by the compressor, this observation suggests that a more effective heat recovery strategy could be implemented to enhance the performance of the system.

A further issue related to thermal management also emerges from the analysis. By examining Figure 3.15, which reports the temperature inside the thermal energy storage during the discharge phase together with the turbine exhaust temperature, it becomes clear that the temperature reached by the TES is not sufficient to prevent icing related problems. Such low temperatures may lead to the formation of ice within the expansion line, potentially causing damage to the turbine. This issue is present from the very beginning of the discharge phase and becomes progressively more severe as the process continues. Since the TES is modelled as a perfectly mixed system, its temperature decreases during discharge, and consequently the turbine inlet temperature drops steadily over time.

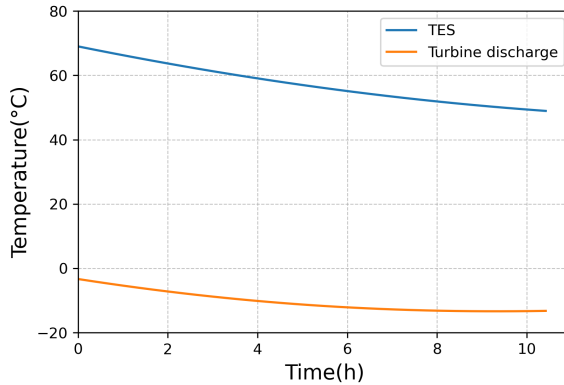


Figure 3.15: Temperature in the thermal storage and at the turbine outlet during a fully discharge phase

### 3.7 Conclusion

This chapter has presented the methodology adopted for the development of a dynamic model of a micro-CAES system and the validation procedures carried out. The model was implemented using OpenModelica, an open-source simulation environment based on the Modelica language, whose acausal, equation-based formulation was shown to be particularly well suited for describing time-dependent energy systems such as CAES.

The developed model represents an existing industrial micro-CAES prototype and a detailed description of the governing equations and component models has been provided. This prototype layout is intentionally adopted as a baseline reference configuration, as it is supported by an extensive experimental dataset that enables a thorough validation of the numerical framework. The model was therefore validated by comparing numerical results with experimental measurements obtained from a previous testing campaign performed on the industrial system.

Once validated, the model was employed to perform a component-level exergetic analysis aimed at identifying and quantifying the dominant loss mechanisms in the reference layout. The results show that the main contributions to inefficiency are associated with compression inefficiency and heat rejection, as well as with exergy destruction related to pressure regulation along the discharge line, which is inherent to the current prototype and pressure-level strategy. The analysis also revealed an additional issue associated with the risk of icing at the turbine exhaust, caused by insufficient pre-heating of the air before expansion, further confirming the limitations of the existing heat-management approach.

Overall, the key outcome of this chapter is the establishment of a validated modeling basis and a quantitative loss breakdown for the baseline prototype, which will be used in the following chapters to investigate alternative, better-performing micro-CAES layouts.

## Chapter 4

# Development of a Sizing Method through Sensitivity Analysis and Surrogate-Based Optimization

In this chapter, a revised configuration of the system is proposed, the modifications introduced here are motivated by both the insights gained from that earlier analysis and relevant findings from the literature. An updated thermodynamic model is first presented, followed by the corresponding economic model that will serve as a reference framework for the remainder of the dissertation. The chapter then introduces the sizing procedure, which consists of an initial sensitivity analysis and a subsequent optimization stage. Finally, a case study is provided to demonstrate the functioning and effectiveness of the proposed procedure.

### 4.1 System description

The system under analysis, schematically represented in Figure 4.1, consists of an adiabatic micro-CAES (with a power rating in the order of tens of kW and a storage capacity of hundreds of kWh) coupled with a PV energy source and an end user. The CAES system is composed of a 3-stages reciprocating compressor, six double-pipe heat exchangers, a

stratified thermal energy storage system (TES), a series of high-pressure vessels, a lamination valve and a series of three volumetric expanders. The compression phase is intercooled through three heat exchangers that release heat to the oil contained in a stratified reservoir that stores the thermal energy and supplies it back to the air during the discharge phase, and heat sinks that dissipate heat to the ambient. The air is stored in isochoric vessels, causing the compressor outlet pressure to vary based on vessels pressure, while the lamination valve ensures that the pressure at the inlet of first expander is not higher than a maximum allowable value. The compressor is driven by an electric motor with variable speed, powered by a series of photovoltaic panels, while the expanders move an electric generator that provides power to the users; more details about the load profile are given in the following paragraphs.

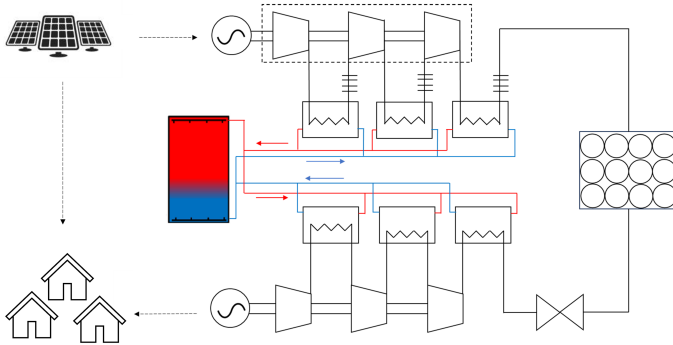


Figure 4.1: Plant layout

The performance limitations of the configuration examined in the previous chapter made it necessary to identify a new plant layout, with the specific aims of increasing overall system efficiency and avoiding the icing issues previously encountered. The configuration proposed in this chapter is designed to address these objectives, and exhibits substantial differences with respect to the previous one, particularly regarding the

machines employed in the compression and expansion phases, as well as the thermal recovery system. Specifically, compression is still performed by a reciprocating compressor, but arranged in three stages rather than four; three scroll expanders are used in place of the micro-turbine; and the thermal recovery system is based on a stratified storage tank with external heat exchangers, rather than immersed ones. The modifications introduced with respect to the previous design are primarily informed by insights drawn from the scientific literature, rather than from a detailed comparative analysis of multiple alternative configurations, since the topic has already been extensively investigated in the literature. The rationale behind each of these choices is detailed in the following sections.

It is necessary to underline that the simplicity of the plant configuration and the exclusive use of commercially available components were overarching constraints of this work. These constraints dictated the choice to rely solely on standard technologies, rather than hypothesizing the adoption of innovative machines that are not yet fully mature such as isothermal compressors and expanders. A further constraint of the study is the assumption that the CAES system has electricity as its only input and output, and is therefore not integrated with waste-heat sources nor designed to provide useful heat or cooling as an additional output.

Finally, it is worth noting that the system is not designed to guarantee 100% renewable energy coverage. A fraction of the energy demand therefore remains unmet by both the photovoltaic system and the CAES, requiring an additional energy source, such as a backup diesel generator or a grid connection. However, the analysis of how this residual demand is supplied lies outside the scope of the present section and will be addressed in subsequent chapters. The central aim of this chapter is to introduce a methodology for the sizing of the proposed system, which is discussed in detail in Sections 4.4.2 and 4.4.3. The approach is based on parametrising the dimensions of each component as a function of one or more characteristic parameter. A sensitivity analysis is first carried out in order to identify the most influential parameters and discard those with negligible impact on system performance. The retained parameters



are then used as decision variables in an optimisation procedure based on Bayesian methods.

#### 4.1.1 Machine selection

Regarding the compression stage, as reported by Zheng et al. [26] and illustrated in Figure 4.2, which provides a general indication for selecting the appropriate compressor type, several technologies are typically employed in Micro CAES systems, including scroll, radial, and piston compressors. Given the need for flexibility in Micro CAES applications, the compressor must remain compact and lightweight. Moreover, higher compression ratios enable greater power capability and improved energy density. Taking into account manufacturability and the comparatively higher achievable pressure ratios, piston compressors emerge as the most suitable solution. Consequently, a reciprocating compressor is retained in this updated plant configuration as well.

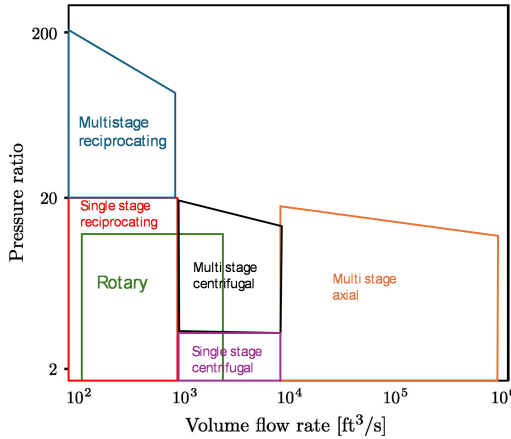


Figure 4.2: Typical operational ranges of different compressor types [161]

A different approach is adopted for the expansion stage, where rotary machines (i.e. scroll and screw expanders) are preferred. For micro-scale applications, volumetric expanders generally offer the best performance,

with scroll and screw types being the most suitable options. However, screw expanders are not recommended for capacities below 10 kW due to leakage losses and manufacturing constraints [112]. At lower power levels, scroll expanders therefore represent the more appropriate choice. Since in this work parametric analysis are performed, in which the expander size may vary, a unified model is used, as a first approximation, to represent both technologies, as will be discussed in more detail in the next section. This assumption is consistent with the results of Dumont et al. [162], who demonstrated that scroll and screw expanders display comparable efficiency curves. Market availability further reinforces the selection of scroll expanders. Piston expanders, for instance, are not commonly offered as commercial standalone units. Scroll machines, by contrast, are widely accessible, moreover a scroll compressor can be readily converted into an expander. This conversion process is well established and relatively straightforward [163, 164], whereas reciprocating compressors do not lend themselves to the same level of adaptability. Piston expanders have nonetheless been investigated in the literature [165, 166], often adapted from internal combustion engines or from HVAC compression devices. Despite this, they tend to suffer from substantial leakage losses, require lubrication and they are characterized by a complex mechanical architecture. In contrast, scroll expanders, owing to their simple geometry and limited number of moving parts, exhibit smaller leakage paths and more stable operation. Their widespread use in HVAC systems and their recent application in Micro CAES configurations [105, 167] make them a more practical and reliable solution for the present study.

As illustrated in Figure 4.1, the proposed configuration employs three compression stages and three expansion stages, with intercooling between the compression stages and heating prior to expansion. The choice of adopting three compression stages is informed by the work of Mohamad et al. [141], one of the earliest studies to introduce the concept of Micro-CAES. Their study presents an extensive parametric investigation on several system design variables. In particular, it examines the influence of the number of compression stages and shows that, for micro-scale systems

with maximum pressures below 200 bar, a three-stage configuration offers an optimal trade-off between efficiency and system complexity.

Finally, the throttling valve is retained to limit the maximum inlet pressure to the expanders. This upper limit is set at 45 bar, consistent with the highest pressure levels achievable by state-of-the-art machines and in agreement with data available from manufacturers [162, 168].

#### 4.1.2 Thermal storage selection

With regard to the thermal recovery system, a solution based on a stratified tank containing mineral oil and equipped with counter-flow double-pipe heat exchangers has been selected. This configuration has the potential to improve heat recovery and potentially overcome the icing issues identified in the previous section while maintaining a relatively low cost. The use of a stratified TES system significantly enhances thermal recovery, as demonstrated by the study of Manente et al. [169], in which various TES configurations for waste-heat recovery and subsequent power generation (via Rankine, Kalina, and steam Rankine cycles) are compared. Their results show that employing a stratified thermal storage system, either liquid or packed-bed, substantially improves recovery efficiency, enabling an energy output approximately 60–70% higher than that obtained with a perfectly mixed thermal store, with the packed-bed configuration performing slightly better overall. Furthermore, Xue and White [170] compare several TES solutions specifically applied to compressed air energy storage systems. In particular, they examine a configuration based on two liquid-thermal-fluid tanks at different temperatures and another based on a packed-bed thermal store, evaluating both options through a techno-economic analysis. Their study concludes that the optimal solution is the adoption of liquid-based TES, since the use of packed beds at high pressure leads to a substantial increase in cost. They also propose a hybrid solution, combining packed-bed storage for the low-pressure stages with liquid storage for the high-pressure stages, which yields a modest performance improvement. However, this option has been dismissed in the present work in order to maintain a simple plant architecture and to

preserve consistency among stages. Their study also compares several heat-transfer fluids for liquid TES systems, identifying mineral oil as the most suitable option. Based on these considerations, a stratified tank has been selected. The use of a single stratified tank, rather than two tanks at different temperature levels, allows for a reduction in both capital cost and space requirements. Heat transfer between the air stream and the mineral oil is achieved through counter-flow double-pipe heat exchangers. This technology offers, on average, high thermal effectiveness, owing to the well defined counter current flow configuration. Moreover, double-pipe exchangers are particularly well suited for micro-scale applications, as reported by [26] thanks to compactness, straightforward manufacturability, and lower cost over larger and more complex shell-and-tube solutions.

## 4.2 Thermodynamic model

As described in Section 3.2 of the previous chapter, the system model was developed using the open-source software OpenModelica. This tool was selected for its ability to simulate time-dependent systems with flexibility and numerical robustness. The model accounts for the dynamic and off-design behaviour of each component to ensure an accurate representation of the overall plant. Component sub-models are parameterized according to their characteristic dimensions, enabling sensitivity analyses with respect to component sizing. The adopted sub-models represent generic components, implemented using state-of-the-art correlations in order to keep both the formulation and the applicability of the modelling procedure as general as possible. This section describes only those components for which a different modelling approach is employed compared to that introduced in the previous chapter. Components whose modelling remains unchanged are not discussed here for the sake of brevity; the reader is referred to the detailed description provided in Section 3.3.

#### 4.2.1 Expander

If ambient heat losses are neglected, scroll and screw expanders can be modelled by means of their isentropic and volumetric efficiencies [164]. The volumetric efficiency is expressed as:

$$\eta_{vol,ex} = \frac{\rho_{in} \cdot \dot{V}_s}{\dot{m}} \quad (4.1)$$

$$\dot{V}_s = V_s \cdot \frac{N}{60} \quad (4.2)$$

where  $\rho_{in}$  is the density at suction conditions, and  $\dot{V}_s$  is the theoretical volumetric flow rate associated with the swept volume of the expander as defined in Equation (4.2). Since the machine operates on a volumetric basis, the theoretical volumetric flow rate is equal to the product of the rotational speed  $N$  and the swept volume  $V_s$ .

The isentropic efficiency is calculated as:

$$\eta_{is} = \frac{\dot{W}}{\dot{m} \cdot w_{is,ex}} \quad (4.3)$$

where  $\dot{W}$  is the mechanical power delivered at the shaft and  $w_{is,ex}$  is the specific isentropic expansion work, defined as in Equation (4.4):

$$w_{is,ex} = \frac{\gamma}{\gamma - 1} \cdot R \cdot T_{in} \cdot \left( \frac{1}{\beta^{\frac{\gamma}{\gamma-1}}} - 1 \right) \quad (4.4)$$

where  $\beta$  is the pressure ratio across the expander.

Both the isentropic and volumetric efficiencies are expressed as functions of the pressure ratio and the rotational speed.

The volumetric efficiency adopted in the present work is derived from the experimental data reported by Zhang et al. [171], who present an experimental characterization of a scroll expander driven by compressed air. The trend of the volumetric efficiency obtained from this analysis is shown in Figure 4.3b. It is worth noting that this efficiency is primarily a function of the rotational speed, whereas the influence of the pressure ratio is comparatively limited. This behavior can be explained by the

intrinsic features of scroll machines, which exhibit a negligible dead volume compared to reciprocating expanders, and by the fact that higher rotational speeds reduce the impact of internal leakage.

The isentropic efficiency, on the other hand, is modelled through an analytical formulation, specifically the one proposed by Declaye et al. [164]. This formulation expresses the isentropic efficiency as a function of the pressure ratio and the rotational speed, and is detailed in the following paragraphs through Equations (4.5) to (4.12). The general expression for the isentropic efficiency is given by:

$$\eta_{is} = \eta_{max} \cdot \sin(\xi \cdot \arctan(B \cdot (\beta - \beta_0) - E \cdot (B \cdot (\beta - \beta_0) - \arctan(B \cdot (\beta - \beta_0)))))) \quad (4.5)$$

where

$$B = \frac{\delta}{\xi \cdot \eta_{max}} \quad (4.6)$$

$$E = \frac{B \cdot (\beta - \beta_0 - \tan(\frac{\pi}{2\xi}))}{B \cdot (\beta_{max} - \beta_0 - \arctan(B \cdot (\beta_{max} - \beta_0)))} \quad (4.7)$$

To account for the influence of rotational speed, the model parameters are corrected as functions of the non-dimensional rotational speed  $N^*$ . Specifically, they are expressed through linear regressions in  $N^*$  using empirical coefficients, as described by Equations (4.8) to (4.12), where the subscript  $n$  indicate nominal conditions.

$$N^* = \frac{N - N_n}{N_n} \quad (4.8)$$

$$\beta_0 = \beta_{0,n} + a_0 \cdot N^* \quad (4.9)$$

$$\delta = \delta_n + a_1 \cdot N^* \quad (4.10)$$

$$\beta_{max} = \beta_{max,n} + a_2 \cdot N^* \quad (4.11)$$

$$\eta_{\max} = \eta_{\max,n} + a_3 \cdot (N^* - N_n^*)^2 \quad (4.12)$$

The values of the empirical coefficients are taken from the reference study. A maximum isentropic efficiency of 70% is assumed, representative of an average commercial expander. This value is not expected to affect the qualitative trends of the results while still providing a realistic estimate, including the economic and environmental outcomes reported in the following chapters. Preliminary analyses were carried out to assess the sensitivity of the model to this parameter and supported this assumption; for the sake of conciseness, these analyses are not reported here, and the influence of this parameter is not further investigated. The maximum

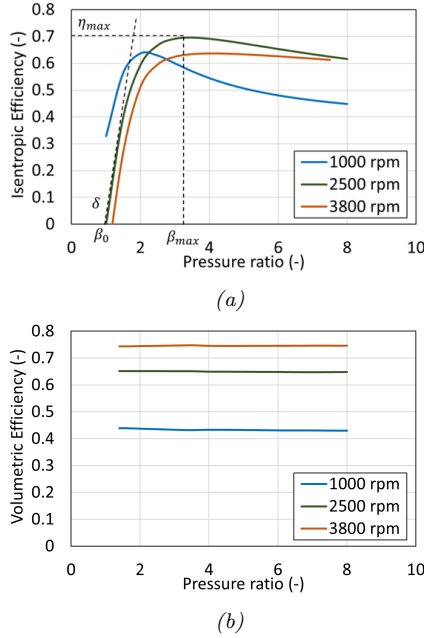


Figure 4.3: Expander efficiency curves as function of pressure ratio and rotational speed: (a) Isentropic efficiency, (b) Volumetric efficiency

efficiency is assumed to remain constant also in the parametric analyses where the machine size is varied. This assumption is justified by the fact that, for volumetric machines, the maximum efficiency is only weakly dependent on the machine size [29], which is one of the main reasons why such machines are particularly suitable for small-scale applications compared to turbomachinery. The resulting isentropic efficiency curve is shown in Figure 4.3a, which also provides a graphical interpretation of the coefficients appearing in the analytical formulation.

### 4.2.2 Compressor

With regard to the compressor, the modelling approach follows the same general methodology described in Section 3.3.3 of the previous chapter, where each compression stage is represented through characteristic curves for both isentropic and volumetric efficiencies, expressed as functions of the pressure ratio and shaft rotational speed. Few modifications are introduced, in particular:

- The outlet temperature is not computed by modelling a heat transfer process; instead, a maximum allowable temperature is imposed. The compressor outlet temperature is therefore taken as the minimum between the adiabatic outlet temperature and a prescribed upper limit, as expressed in Equation (4.13). The thermal energy corresponding to the difference between the adiabatic and the outlet temperatures is assumed to be dissipated to the surroundings. A maximum temperature of 150°C is adopted to mimic the typical constraints of standard industrial machines.

$$T_{\text{out}} = \min(T_{\text{ad}}, T_{\text{max}}) \quad (4.13)$$

- The isentropic efficiency is modelled using the same approach introduced in the previous paragraph for the expander. The coefficients of the corresponding correlation have been obtained by fitting in-house experimental data and are reported in Table 4.1.

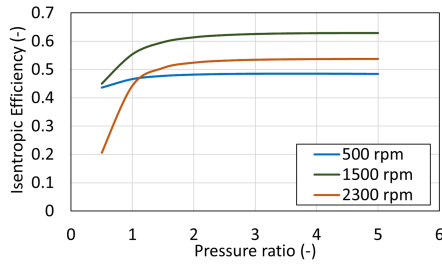


Table 4.1: Coefficients of the analytical correlation adopted for the compressor isentropic efficiency

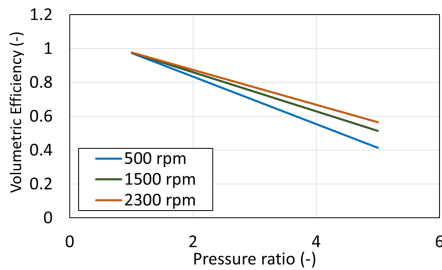
| $\beta_{0,n}$ | $\delta_n$ | $\xi$ | $\beta_{max,n}$ | $N_n$ | $\eta_{max,n}$ |
|---------------|------------|-------|-----------------|-------|----------------|
| -0.1          | 0.886      | 1.617 | 5               | 1500  | 0.63           |

- The general curves for volumetric efficiency are derived from Lyons and Plisga [172].

Figure 4.4 shows the resulting isentropic and volumetric efficiency curves as functions of the compression ratio for selected values of rotational speed.



(a)



(b)

Figure 4.4: Compressor efficiency curves as function of pressure ratio and rotational speed: (a) Isentropic efficiency, (b) Volumetric efficiency

### 4.2.3 Thermal energy storage

Regarding the TES, stratification is modeled by considering a cylinder and slicing it along its axis into a series of control volumes at uniform temperature. For each of these volumes, the energy balance equation was applied, as reported in Equation (4.14), accounting for mass transport, conductive heat exchange between layers and heat exchange with the external environment. The buoyancy effect was not included in the modeling, as it represents a limited contribution to the system's stratification in this configuration, where the latter is primarily ensured by introducing hot oil at the top of the reservoir and cold oil at the bottom. The heat exchange with the external environment is calculated for each layer as the one between two sources at fixed temperature connected through a series of three thermal resistances: the first represents the convection between the thermal fluid and the TES wall, the second the conduction through a cylindrical wall, while the third accounts for natural convection on a vertical cylindrical surface. The tank is assumed to be cylindrical in shape with an aspect ratio of 4, a value commonly referenced in the literature to prevent the use of excessively elongated tanks [173].

Figure 4.5 graphically shows the applied schematization of the TES.

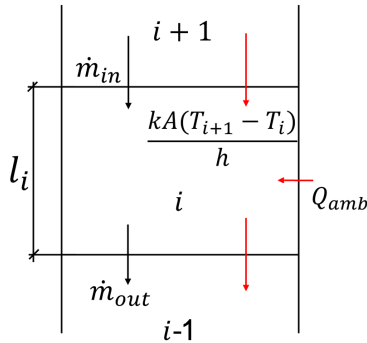


Figure 4.5: Schematic representation of a TES layer

$$\rho c_p V_i \frac{dT_i}{dt} = \dot{m}_{in} \cdot h_{in} - \dot{m}_{out} \cdot h_{out} + \frac{kA}{l_i} (T_{i+1} - 2T_i + T_{i-1}) + \dot{Q}_{amb} \quad (4.14)$$

#### 4.2.4 Heat exchanger

The double-pipe heat exchangers are modelled using the  $\varepsilon$ -NTU approach (Equations (4.15) to (4.17)), introducing a dependence of the global heat-transfer coefficient on the mass flow rate as described by Equation (4.18). In this expression, the subscript  $n$  denotes nominal conditions (i.e., the mass-flow rate corresponding to machines operating at their nominal speed and pressure ratio). The correlation proposed in [174] is employed to account for variations in the convective heat-transfer coefficients induced by changes in mass flow rate. The exponent associated with the mass flow term reflects that of the Reynolds number in the Dittus–Boelter correlation for turbulent flow in a tube.

For the sizing of the heat exchangers, a design heat transfer coefficient of  $U_n = 250, \text{ W}/(\text{m}^2, \text{ K})$  is assumed, based on typical values reported in [175]. As a first approximation, the convective coefficients at design point of air and oil side ( $\alpha_{\text{air},n}$  and  $\alpha_{\text{oil},n}$ ) are assumed to be equal. The required heat transfer surface is then computed for each system configuration as the area ensuring the prescribed NTU value, given the mass flow rates handled by the compressor and expander under nominal operating conditions. The oil mass flow rate within the heat exchanger is set to 0.55 times that of the air. This value, corresponding to the ratio between the specific heats of air and mineral oil at ambient temperature, is chosen to maintain a thermal capacity ratio close to unity, thereby maximising the effectiveness of thermal recovery.

$$\dot{m} \cdot (h_{in} - h_{out}) = \varepsilon \cdot \dot{Q}_{max} \quad (4.15)$$

$$\varepsilon = \frac{1 - e^{(-NTU \cdot (1-Cr))}}{1 - Cr \cdot e^{(-NTU \cdot (1-Cr))}} \quad (4.16)$$

$$NTU = \frac{UA}{C_{min}} \quad (4.17)$$

$$\frac{1}{U} = \frac{1}{\alpha_{air,n}} \left( \frac{\dot{m}_{air,n}}{\dot{m}_{air}} \right)^{0.8} + \frac{1}{\alpha_{oil,n}} \left( \frac{\dot{m}_{oil,n}}{\dot{m}_{oil}} \right)^{0.8} \quad (4.18)$$

### 4.2.5 Integration with renewable generation and load

The CAES system under study is connected to a photovoltaic (PV) source and an electrical load, consistently with the configurations introduced in the previous sections. The temporal evolution of the system is driven by the power output of the PV plant and by the user demand.

#### 4.2.5.1 Photovoltaic Power Model

The photovoltaic generation is modelled through a simplified formulation in which the instantaneous output power is computed as

$$P_{PV}(t) = PR \cdot \frac{I(t)}{I_{STC}} \cdot P_{PV,p}, \quad (4.19)$$

where  $I(t)$  is the in-plane irradiance,  $I_{STC} = 1000 \text{ W/m}^2$  is the reference irradiance at standard test conditions,  $P_{PV,p}$  is the installed peak power, and  $PR$  is the performance ratio of the photovoltaic system, namely an overall system efficiency that accounts for the cumulative effect of all loss mechanisms (module temperature, soiling, wiring, mismatch, inverter efficiency, etc.), consistent with the definition given by the IEC 61724-1 standard [176]. The time-varying solar irradiance  $I(t)$  is computed by means of the **PhotoVoltaics** library in OpenModelica [177], which, using the approach proposed by Quaschnig [178], analytically evaluates the irradiance as a function of the geographical location, date, time of day, and PV module parameters (such as tilt angle and orientation). The model adopts a clear-sky formulation, neglecting atmospheric attenuation, cloud cover and diffuse radiation, and therefore provides a deterministic upper-bound estimate of the available irradiance.

#### 4.2.5.2 Load Profile

The user demand is prescribed through a time-dependent load profile:

$$P_L(t) \quad \forall t \in [0, T], \quad (4.20)$$

which is provided as an external input. Further details on the specific case study are presented in Section 4.4.4.

#### 4.2.5.3 Compressor and Expander Operating Logic

The compressor and the expander regulate their delivered (or absorbed) power by modulating their rotational speed  $N(t)$  within the admissible range:

$$N_{\min} \leq N(t) \leq N_{\max}. \quad (4.21)$$

Their objective is to track the instantaneous power difference between PV production and load demand:

$$\Delta P(t) = P_{PV}(t) - P_L(t), \quad (4.22)$$

within the limits imposed by the maximum allowable pressure in the storage vessel and the characteristic performance curves.

Based on the value of this power difference and the internal tank pressure, the system can operate in charging mode (compression), discharging mode (expansion), or in an idle state. Specifically:

- **Charging Phase**

The system enters the charging phase when the power surplus is sufficient to activate the compressor and the tank pressure is below its maximum limit:

$$\Delta P(t) \geq P_{\min}^{\text{comp}}, \quad (4.23)$$

$$p(t) < p_{\max}. \quad (4.24)$$

The quantity  $P_{\min}^{\text{comp}}$  represents the minimum power required to operate the compressor at the minimum admissible rotational speed

$N_{\min}$ ; otherwise, the system remains idle even in the presence of a positive but insufficient surplus. When the compressor is active, the absorbed power is imposed as the minimum between the available power surplus and the maximum admissible compressor power at maximum rotational speed, consistently with the current stage pressure ratios:

$$P_{\text{comp}}(t) = \min [\Delta P(t), P_{\text{comp}}^{\max}(N_{\max}, \beta_i)] . \quad (4.25)$$

The operating point  $(\beta_i, N_i)$  for each compression stage is then iteratively determined from the compressor maps, such that the resulting absorbed power matches the imposed value  $P_{\text{comp}}(t)$ .

#### • Discharging Phase

A discharging phase occurs when PV generation is insufficient to meet the load and the tank pressure is above the minimum operating limit:

$$\Delta P(t) < 0, \quad (4.26)$$

$$p(t) > p_{\min}. \quad (4.27)$$

A minimum deliverable power  $P_{\min}^{\text{exp}}$  corresponds to the operation of the expander at its minimum rotational speed  $N_{\min}$ . Similarly, the maximum power output allowed by the expander characteristics is indicated as  $P_{\text{exp}}^{\max}(t)$ .

The expander power is computed as:

$$P_{\text{exp}}(t) = \min [\max(-\Delta P(t), P_{\min}^{\text{exp}}), P_{\text{exp}}^{\max}(t)] . \quad (4.28)$$

If the required power is below the minimum admissible level, the expander must still operate at  $N_{\min}$ ; in such cases, the excess mechanical power is dissipated.

• **Idle State**

If neither the charging nor the discharging conditions are satisfied, the system remains idle:

$$P_{\text{comp}}(t) = P_{\text{exp}}(t) = 0. \quad (4.29)$$

The complete control logic is summarised in Figure 4.6.

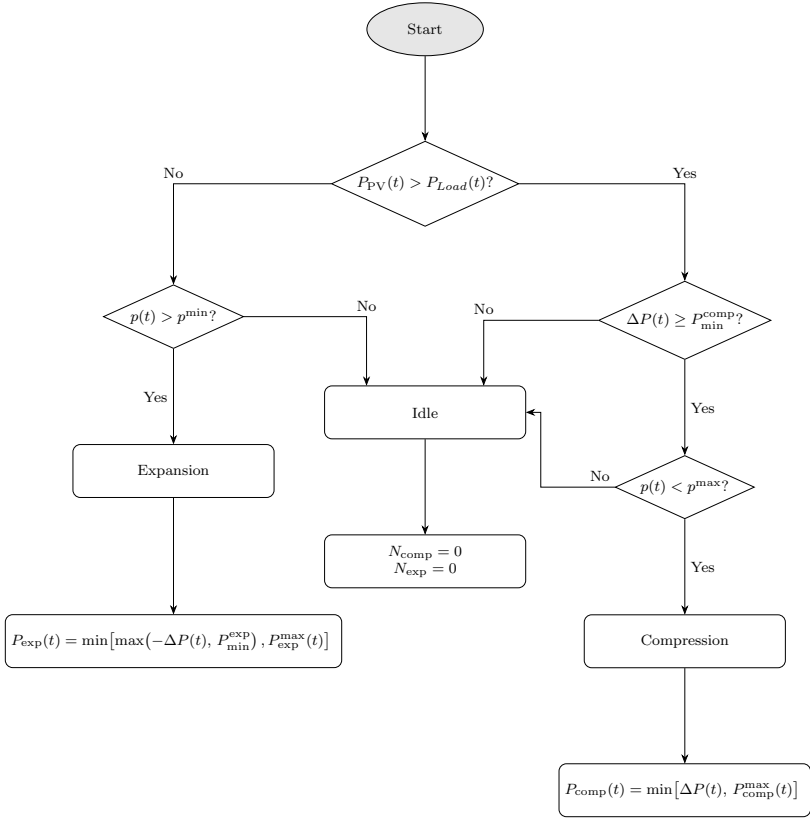


Figure 4.6: Control logic of the PV-CAES system

### 4.2.6 Energy performance indicators

Two performance indicators are adopted to assess the energy performance of the system: the round-trip efficiency and the energy output. The RTE, already introduced and defined as the ratio between the energy produced during the expansion phase and the energy absorbed by the compressor over a reference period (see Equation (3.22)), is evaluated here on an annual basis. Differently from the previous chapter, it is not computed over a single complete charge–discharge cycle, but rather based on the total energy absorbed and generated over one year of operation. This definition allows the RTE to account for operational variability and for the effects of off-design operation.

In addition, the absolute amount of energy delivered by the CAES system to the load is considered as a key performance indicator in the subsequent analyses. When varying the size of the different system components, it is essential not only to evaluate the system efficiency, but also to quantify the amount of energy that is effectively produced and supplied by the CAES system.

## 4.3 Economic analysis

To analyse the system from an economic perspective, the Levelized Cost of Storage (LCOS) is adopted as the primary performance metric. LCOS, defined as in Equation (4.30), is a widely accepted indicator in the literature for the economic assessment of energy storage technologies [145]. It represents the average cost per unit of energy discharged over the entire lifetime of the storage system. In Equation (4.30), CAPEX denotes the total investment cost of the system, while O&M represents the annual operation and maintenance expenses. The term  $C_{ch}$  accounts for the cost associated with the charging phase of the storage system and is calculated as the product of the electricity price  $c_{el}$  and the annual electrical energy input  $E_{in}$ . The quantity  $E_{out}$  refers to the total useful energy delivered by the storage system over its entire lifetime. Finally,  $d$  denotes the discount rate and  $Y$  the total number of years considered in



the system lifetime. A system lifetime of 25 years is assumed. Annual operation and maintenance costs are taken as 2% of the total capital investment, while the discount rate is fixed at 5%. These assumptions are consistent with values commonly adopted in techno-economic analyses of comparable energy storage technologies [179–181].

The charging cost is evaluated by computing the Levelized Cost of Electricity (LCOE) (Equation (4.32)) of the energy produced by the photovoltaic (PV) system, considering only the useful electricity, namely the portion either directly supplied to the load or used by the CAES during the charging phase. Electricity that is generated but neither consumed nor stored is excluded from the LCOE calculation; otherwise, the resulting LCOE would be artificially reduced and would not correctly reflect the actual cost of the energy effectively utilised by the system. It is worth noting that the LCOS formulation is conceptually analogous to that of the LCOE. The two metrics share the same structure in terms of discounted costs and energy output over the system lifetime; however, in the LCOE formulation the charging cost does not appear.

This methodological approach is adopted because the primary objective of the analysis is to identify a potentially optimal system configuration, as a trade-off between the considered energy performance metrics, rather than to assess the absolute economic feasibility of the system. As already mentioned, the proposed system is not intended to provide full renewable energy coverage, and a residual share of the demand must be supplied by an auxiliary source. The detailed analysis of this aspect, together with a comprehensive assessment of the overall economic viability of the system, is deferred to subsequent chapters.

$$LCOS = \frac{CAPEX + \sum_{i=1}^N \frac{O\&M}{(1+d)^i} + \sum_{i=1}^N \frac{C_{ch}}{(1+d)^i}}{\sum_{i=1}^N \frac{E_{ex}}{(1+d)^i}} \quad (4.30)$$

$$C_{ch} = c_{el} \cdot E_{in} = LCOE_{PV} \cdot E_{in} \quad (4.31)$$

$$LCOE = \frac{CAPEX + \sum_{i=1}^N \frac{O\&M}{(1+d)^i}}{\sum_{i=1}^N \frac{E_{ex}}{(1+d)^i}} \quad (4.32)$$

### 4.3.1 Capital cost estimation

In order to estimate the investment cost of each component, cost correlations available in the literature are employed, each expressed as a function of the key characteristic parameters of the respective component. These correlations are here summarized.

- Compressor

As discussed in [182], the cost of mechanical equipment can often be expressed as a power law function of a characteristic design parameter, as shown in Equation (4.33).

$$C_A = C_{A0} \cdot \left( \frac{A}{A_0} \right)^z \quad (4.33)$$

where  $C_0$  denotes the reference capital cost,  $A_0$  the reference capacity, and  $z$  ( $z < 1$ ) is the cost exponent, which depends on the specific equipment type. For the compressor, this scaling approach is adopted using the mechanical power (expressed in kW) as the characteristic parameter. The coefficients, derived from [45], are:

$$C_0 = 73634\$, \quad A_0 = 75kW, \quad z = 0.84.$$

- Expander

As for the compressor, the capital cost of volumetric expander is function of its power and reported by [183]

$$CAPEX_{expander} = 1810 \cdot P^{0.71} \quad (4.34)$$

- Heat exchanger:

The cost of double-pipe heat exchangers is estimated using the correlation provided by Turton et al. [182]. In this formulation, the base cost is expressed as a function of the heat-transfer area and subsequently corrected for the operating pressure through the pressure factor  $F_p$  (Equations (4.35) to (4.37)). The values assumed by the coefficients appearing in the definition of  $F_{p_{hex}}$  are summarized in Table 4.2

$$CAPEX_{hex}(\$) = CAPEX_{hex,0} \cdot F_p \quad (4.35)$$

$$CAPEX_{hex,0} = 10^{3.344} \cdot A^{0.2745} \cdot e^{-0.04722 \cdot \frac{\ln(A)}{\ln(10)}} \quad (4.36)$$

$$F_{p_{hex}} = 10^{k_{p1}} \cdot p^{k_{p2}} \cdot e^{k_{p3} \cdot \frac{(\ln(p))^2}{\ln(10)}} \quad (4.37)$$

Table 4.2: Coefficient for pressure factor calculation for heat exchanger

| $k_{p1}$ | $k_{p2}$ | $k_{p3}$ | Pressure (bar) |
|----------|----------|----------|----------------|
| 0        | 0        | 0        | < 5            |
| -0.00164 | -0.00627 | 0.0123   | [5; 140]       |
| 13.1467  | -12.6574 | 3.0705   | [140; 300]     |

- Photovoltaic system:

The cost of the photovoltaic system is derived from the LCOE Report – 2023 published by Lazard [184], which accounts for both PV modules and power conversion equipment (e.g., inverters). The cost is expressed as a function of the installed peak power capacity.

$$CAPEX_{PV}(\$) = 1050 \cdot P_p(kW_p) \quad (4.38)$$

- Electric motor and generator

For the electric motor and generator, the same cost-scaling approach adopted for the compressor is applied. The capital expenditure is estimated using a power-law correlation of the form:

$$CAPEX_{\text{motor/generator}} = 725 \cdot \left( \frac{P \text{ (kW)}}{10} \right)^{0.92}, \quad (4.39)$$

where the reference cost and reference power values derived from industrial experience, while the scaling exponent is taken from [185].

- Thermal storage

Both the cost of the atmospheric storage tank and that of the thermal oil are estimated using the correlations provided in [170]. In particular, the tank cost is expressed as a function of its volume (Equation (4.40)), while the oil cost is evaluated based on the total mass of fluid stored within the TES system (Equation (4.41)).

$$CAPEX_{TES}(\$) = 500 \cdot V \text{ (m}^3\text{)} \quad (4.40)$$

$$CAPEX_{oil}(\$) = 2.7 \cdot m \text{ (kg)} \quad (4.41)$$

- Pressure vessels

As discussed in detail by [45], it is generally more advantageous to store compressed air in a large number of small, high-pressure tanks rather than in fewer vessels of larger volume and lower pressure. Moreover, for a fixed storage pressure, the cost decreases as the length-to-radius ratio increases. In this work, standard commercial cylinders with a volume of 50 L and a diameter of 0.229 m are considered.

The cost of the storage vessels is estimated using the correlation proposed by Houssainy et al. [186] and reported in Equation (4.42), based on data from several manufacturers (Fjords Processing, KS

Industries, Modern Custom Fabrication Inc.). Here,  $V$  denotes the total vessel volume. The pressure factor  $Fp_{\text{vessel}}$ , derived from the ASME Boiler and Pressure Vessel Code [182], is defined in Equation (4.43), where  $p_{\text{max}}$  is the maximum operating pressure and  $D$  is the tank diameter.

$$CAPEX_{\text{vessel}} = 1000 \cdot V \text{ (m}^3\text{)} \cdot Fp_{\text{vessel}} \quad (4.42)$$

$$Fp_{\text{vessel}} = \frac{1}{0.0063} \left( \frac{(p_{\text{max}}(\text{bar}) + 1) D(m)}{2(850 - 0.6(p_{\text{max}}(\text{bar}) + 1))} + 0.00315 \right) \quad (4.43)$$

### 4.3.2 Effect of purchase time

When using cost correlations, it is essential to update the estimated costs to present day values in order to account for inflation and other macroeconomic factors that affect price evolution over time. To this end, the cost estimates are adjusted to the current economic context using the Chemical Engineering Plant Cost Index (CEPCI), as shown in Equation (4.44), ensuring that the investment costs reflect contemporary price levels. The CEPCI values for system components reported in Table 4.3 are taken from [187].

$$Cost_{\text{actual}} = Cost_{\text{ref}} \cdot \frac{CEPCI_{\text{actual}}}{CEPCI_{\text{ref}}} \quad (4.44)$$

Table 4.3: CEPCI index for main system components

| Component      | Year | CEPCI | Component        | Year | CEPCI |
|----------------|------|-------|------------------|------|-------|
| Compressor     | 2022 | 816   | Thermal oil tank | 2018 | 605.2 |
| Expander       | 2005 | 468.5 | Thermal oil      | 2018 | 605.2 |
| Heat exchanger | 2001 | 394.3 | PV system        | 2023 | 803.3 |
| Air reservoir  | 2022 | 816   | Motor/Generator  | 2018 | 605.2 |

## 4.4 Numerical method

### 4.4.1 Software coupling

Both the sensitivity analysis and the optimization process were carried out by coupling OpenModelica with Dakota, an open-source toolkit that offers a flexible and extensible interface between simulation environments and iterative analysis methods [188]. Dakota provides a broad suite of algorithms for sensitivity and variance analysis, optimization, uncertainty quantification, and various other statistical evaluations.

As noted in [189], a key advantage of Dakota lies in its ability to interface with external simulation tools through a lightweight integration that does not require any detailed knowledge of the internal structure of the simulator. This approach, often referred to as black-box coupling, enables Dakota to operate without accessing or modifying the source code of the computational model. Instead, Dakota and the external simulator communicate by exchanging simple input and output files. The user specifies the analysis settings in a Dakota input file, including the file names associated with the external simulation code. During execution, Dakota automatically launches the simulator as an external process, handles the data exchange, and iteratively manages the analysis workflow.

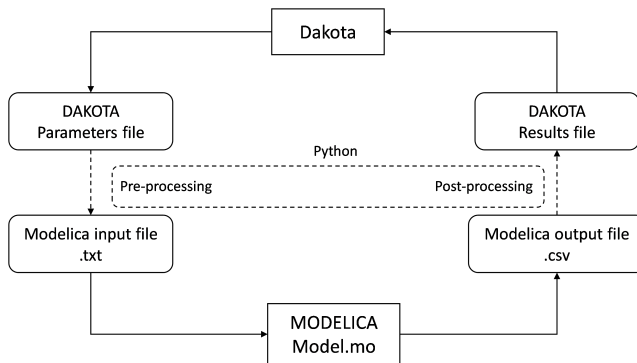


Figure 4.7: Modelica and Dakota coupling workflow

The overall coupling structure is schematically illustrated in Figure 4.7. In this work, the simulation interface was implemented using Python scripts and the OMPython library [190], which provides the official Python API for interacting with OpenModelica.

#### 4.4.2 Sensitivity analysis

The sensitivity analysis aims to identify which system sizing parameters have the greatest influence on the selected performance metrics (RTE,  $E_{out}$  and LCOS), to assess whether certain parameters can be excluded from the optimization process, thereby reducing its computational cost. To evaluate the impact of system sizing, at least one sizing parameter is selected for each major system component, primarily representing its characteristic scale.

Specifically, the parameters considered are:

- the swept volume of the compressor and expander,
- the storage volume and maximum storage pressure,
- the number of transfer units used to size the heat exchangers,
- the scaling parameter employed to size the thermal energy storage.

Indeed, the size of the TES is not varied independently of the other components, as it is intrinsically linked to the amount of heat that must be stored, which in turn depends on the dimensions and operating conditions of the remaining system elements. A preliminary TES sizing is therefore performed based on the total amount of storable air. The thermal storage mass is estimated as the total amount of storage medium that would circulate at a mass flow rate equal to 0.55 times the air mass flow rate through the heat exchangers under the most demanding operating conditions, namely when the air tanks are either fully charged from an empty state or fully discharged from a full state. During the sensitivity analysis, this TES sizing is varied by multiplying this reference value by a dimensionless scaling factor, hereafter denoted as  $M_{mult}$ .

Other parameters describing the CAES thermodynamic cycle (e.g., maximum TES operating temperature, maximum compressor outlet temperature) are not included in the sensitivity analysis. These quantities are strongly dependent on the technology level of individual components; improvements in such parameters would naturally lead to enhanced system performance, but they fall outside the scope of the present work.

The sensitivity analysis is performed by using the method developed by Morris [191] with the correction subsequently proposed by Campolongo et al. [192]. This method allows to rank the parameters influence on the output, and it was selected since it allows to deal with non-monotonic relationship and to consider interaction between variables differently from simpler approach (e.g. method Spearman and Pearson correlation coefficient). At the same time, it guarantees a low computational cost (i.e. number of simulated points) if compared with other global sensitivity methods based on variance decomposition, such as Sobol indexes, at a slightly reduced accuracy [193]. As the goal of this stage is to achieve a quick qualitative comparison between the impact of different parameters, the method was considered well suited for it. This method involves a sampling strategy that allows to compute the elementary effect of each input on the output; the final outputs of the method are the modified mean  $\mu_i^*$  and variance  $\sigma_i$  of the elementary effect. The mean  $\mu_i^*$  is a measure of influence of the  $i$ -th parameter on the output. The greater it is, the more the  $i$ -th input contributes to the dispersion of the output. The variance  $\sigma_i$  is a measure of non-linearity and interaction effects of the input, low value suggests a linear relationship between input and output while a large value will indicate non-linear effects, or interaction with at least one other variable.

#### 4.4.3 Optimization process

Subsequently to the sensitivity analysis, three optimization processes were carried out to identify the plant configuration that optimize the RTE, the output energy and the LCOS separately. The design parameters that



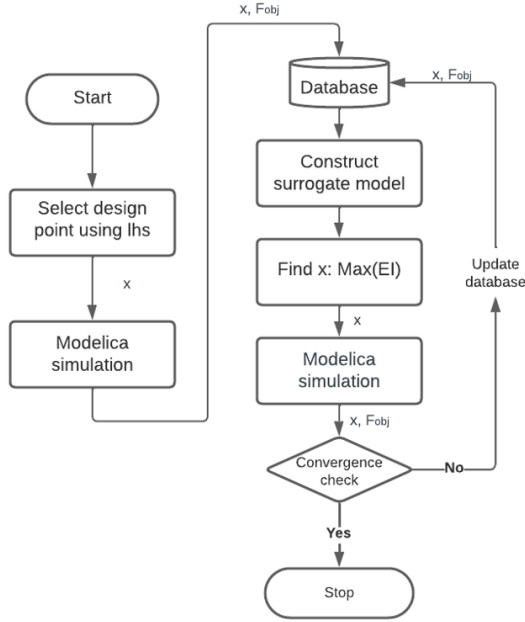


Figure 4.8: Optimization process workflow

are varied during the optimization process are the same as those examined in the sensitivity analysis, except for those identified as negligible in the sensitivity analysis results. The optimization algorithm used to minimize the objective function ( $F_{obj}$ ) is based on the Efficient Global Optimization (EGO) methodology. This method was firstly proposed by Jones et al. [194] to facilitate the minimization of expensive implicit response functions by using a surrogate model of the real one; this method allows to reduce the number of iterations to be carried out compared to other optimization algorithms (e.g. genetic algorithm) [195]. The optimization process is depicted in Figure 4.8 and here briefly described:

1. Firstly, a set of sample point is generated in the design space using the Optimal Latin Hypercube Sampling (OLHS) providing a good representation of the design space [196], the number of initial points is chosen by the algorithm based on the number of variables. For these points the objective function is evaluated through the thermodynamic and economic model creating a starting database.
2. The data available in the database are used to build a Gaussian process (GP) approximation for  $F_{obj}$ , so that it can be approximated by an analytical formulation.[197]
3. The key aspect of the EGO process is to find the point in the design space that maximize an Expected Improvement (EI) function, defined as the expectation that each configuration will provide a better solution than the current best, based on both the expected values and variances computed by the GP model (i.e. points with parameters values that are “far” from the investigated ones will bear a higher uncertainty, thus potentially present a high EI even if  $F_{obj}$  predicted by the surrogate is lower than the current minimum).
4. Modelica analysis is then performed for the parameters that maximize the EI to evaluate the real value of  $F_{obj}$  in this point.
5. Finally, a convergence check is performed at each iteration: if the criterion is satisfied, the loop is terminated; otherwise, the optimization process is continued and the database is updated with the newly computed value to construct an updated surrogate model. The convergence criterion adopted is based on the value of the Expected Improvement. Since the EI is a dimensional quantity, the convergence threshold is defined according to the expected order of magnitude of the objective function. Specifically, convergence is enforced when the EI falls below  $10^{-4}$  times the expected order of magnitude of the objective function. This convergence criterion is consistently adopted in all the optimization studies presented in this thesis.

#### 4.4.4 Test-case set-up

The proposed sizing procedure is tested by considering, as a representative case study, an application installed in Florence, Italy, serving a predefined user demand.

The photovoltaic power generation is modelled according to Equation (4.19) ( $P_{PV}(t) = PR \cdot \frac{I(t)}{I_{STC}} \cdot P_{PV,p}$ ), as discussed in Section 4.2.5.1, based on the instantaneous irradiance and on the performance ratio. As introduced above, the irradiance is computed analytically for the specific location; Figure 4.10 shows the daily irradiance profiles for the selected site on the first day of January and of July. Regarding the performance ratio, for grid-connected crystalline silicon installations, Khalid et al. [198] report values in the range 0.75–0.90; in the present work, a value of 0.75 is adopted, corresponding to the lower bound of the reported range. This conservative choice partially mitigates the optimistic bias of the deterministic clear-sky irradiance model and additionally provides a margin to account for long-term performance degradation, which is not explicitly modelled.

Regarding the required power, a typical domestic load profile is adopted, with a peak demand of approximately 11 kW to represent multiple simultaneous users, as shown in Figure 4.9. The load profile is cyclically repeated for each day of the year. Seasonal variations in demand are neglected, and an average load profile is assumed derived from [199].

Each simulation is performed by running the model over one representative week for four months corresponding to the four seasons (i.e., January, April, July, and October). The simulations are carried out sequentially, with the initial state of each month set equal to the final state of the previous one. A global cyclic operation constraint is imposed by enforcing the equality between the storage pressure at the beginning of the first simulated period and at the end of the last one:

$$p(t_{\text{start}}) = p(t_{\text{end}}) \quad (4.45)$$

It was verified that, with this approach, the results in terms of round-

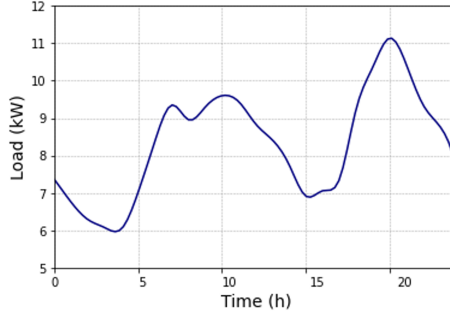


Figure 4.9: User load profile

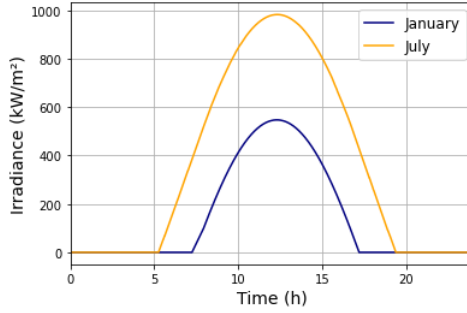


Figure 4.10: Computed global solar irradiance for the test-case location (Florence)

trip efficiency, energy output, and cost are nearly identical to those obtained by simulating the entire year, while drastically reducing the computational time. For illustrative purposes, representative results obtained from a Latin hypercube sample of 20 configurations within the explored design space are reported in Table 4.4, where the round-trip efficiencies and the corresponding relative errors are compared for full-year simulations and for the reduced four-week approach.

The range of variation for each sensitivity parameter is defined based on two preliminary extreme sizing cases. Specifically, the system dimen-

Table 4.4: Representative comparison of round-trip efficiency and relative error between full-year and four-week simulations

| $V_{s,c}$<br>( $\text{cm}^3$ ) | $V_{s,ex}$<br>( $\text{cm}^3$ ) | $V$<br>( $\text{m}^3$ ) | $p_{max}$<br>(bar) | $P_{PV,p}$<br>(kW) | NTU<br>(-) | $m_{mult}$<br>(-) | RTE       |         | $\Delta$<br>(%) |
|--------------------------------|---------------------------------|-------------------------|--------------------|--------------------|------------|-------------------|-----------|---------|-----------------|
|                                |                                 |                         |                    |                    |            |                   | full-year | 4-weeks |                 |
| 2421                           | 46.6                            | 26.6                    | 84                 | 32                 | 2.18       | 0.84              | 33.93     | 33.97   | 0.12            |
| 2864                           | 50.2                            | 20.4                    | 71                 | 150                | 4.12       | 1.89              | 29.16     | 29.19   | 0.09            |
| 1173                           | 34.1                            | 18.1                    | 123                | 108                | 4.47       | 1.13              | 31.12     | 31.17   | 0.14            |
| 1821                           | 17.2                            | 39.7                    | 105                | 96                 | 5.83       | 2.29              | 37.90     | 38.07   | 0.43            |
| 3719                           | 31.2                            | 33.6                    | 112                | 133                | 2.27       | 2.08              | 31.01     | 31.11   | 0.35            |
| 3581                           | 18.9                            | 42.4                    | 135                | 69                 | 5.39       | 2.74              | 42.11     | 42.28   | 0.40            |
| 3994                           | 44.9                            | 24.1                    | 138                | 130                | 4.62       | 1.38              | 27.73     | 27.78   | 0.16            |
| 763                            | 48.8                            | 32.0                    | 182                | 119                | 5.76       | 0.50              | 34.06     | 34.10   | 0.12            |
| 1416                           | 23.0                            | 10.7                    | 187                | 170                | 3.56       | 2.76              | 29.33     | 29.39   | 0.20            |
| 1642                           | 59.7                            | 49.9                    | 166                | 82                 | 4.36       | 2.48              | 30.50     | 30.52   | 0.06            |
| 970                            | 26.7                            | 45.5                    | 49                 | 194                | 3.74       | 1.07              | 35.79     | 35.94   | 0.42            |
| 515                            | 29.8                            | 23.7                    | 95                 | 56                 | 2.41       | 2.13              | 33.08     | 33.13   | 0.15            |
| 2646                           | 54.7                            | 16.5                    | 62                 | 102                | 5.45       | 0.88              | 30.91     | 30.93   | 0.07            |
| 2505                           | 38.4                            | 28.1                    | 200                | 51                 | 3.22       | 1.69              | 36.72     | 36.75   | 0.08            |
| 3166                           | 20.2                            | 13.0                    | 156                | 166                | 4.83       | 1.33              | 30.14     | 30.21   | 0.22            |
| 3366                           | 36.9                            | 37.2                    | 168                | 178                | 2.73       | 0.65              | 29.74     | 29.79   | 0.19            |
| 3108                           | 12.5                            | 40.4                    | 76                 | 45                 | 5.18       | 1.57              | 38.16     | 38.28   | 0.33            |
| 1355                           | 11.7                            | 46.2                    | 144                | 145                | 2.87       | 1.81              | 33.27     | 33.38   | 0.34            |
| 2159                           | 41.5                            | 15.5                    | 45                 | 80                 | 3.86       | 2.92              | 35.49     | 35.54   | 0.12            |
| 1928                           | 56.5                            | 35.8                    | 97                 | 187                | 3.17       | 2.58              | 25.86     | 25.91   | 0.21            |

Table 4.5: Independent sizing parameters and exploration ranges used in the sensitivity analysis

| Parameter                       | Symbol     | Range                        |
|---------------------------------|------------|------------------------------|
| Compressor swept volume         | $V_{s,c}$  | $500 \div 4000 \text{ cm}^3$ |
| Expander swept volume           | $V_{s,ex}$ | $10 \div 60 \text{ cm}^3$    |
| Pressure vessels volume         | $V$        | $10 \div 50 \text{ m}^3$     |
| Thermal oil mass multiplier     | $M_{mult}$ | $0.5 \div 3$                 |
| Max air pressure inside vessels | $p_{max}$  | $40 \div 200 \text{ bar}$    |
| PV installed peak power         | $P_{PV,p}$ | $30 \div 200 \text{ kW}$     |
| NTU at design point             | NTU        | $2 \div 6$                   |

sions required to store sufficient energy to cover a full day at the winter and summer solstices are estimated, considering both the minimum and maximum storage pressure values investigated. The minimum storage pressure is set slightly below the maximum inlet pressure of the expander, thereby avoiding the need for throttling at the expander inlet. Conversely, the maximum storage pressure is selected as a realistic value

for conventional pressure vessels, in order to limit excessive throttling losses, as discussed in the previous chapter. This procedure enables the identification of physically meaningful lower and upper bounds for the system dimensions. The corresponding parameter ranges are summarized in Table 4.5. The swept volume reported in the table refers to the first compression/expansion stage. The volumetric ratios among the different stages are kept constant throughout the analysis and are defined based on a preliminary design assessment.

## 4.5 Results of the sizing procedure

### 4.5.1 Results of the sensitivity analysis

The results of the sensitivity analysis are reported in Figure 4.11 using Morris plots, in which the elementary effects associated with each input parameter are represented in the  $\mu^* - \sigma$  plane for the considered outputs.

Overall, the analysis indicates that the compressor swept volume, the maximum storage pressure, and the reservoir volume are the most influential parameters across all the investigated performance indicators. Conversely, the oil mass multiplier used to size the thermal energy storage and the NTU adopted for the heat exchanger design exhibit the lowest influence. A more detailed inspection of the results shows that, with respect to the RTE, the dominant parameters are the maximum storage pressure and the compressor swept volume. As previously discussed, throttling losses and compressor-related inefficiencies play a crucial role. Therefore, an appropriate selection of both the compressor size and the storage pressure is essential to reduce throttling losses and to ensure compressor operation close to its optimal working regime for extended periods, where efficiency is maximized. A similar consideration applies to the expander displacement, which shows a more pronounced impact on RTE compared to its influence on the other performance metrics.

The total energy output ( $E_{out}$ ) is primarily governed by the overall capacity of the system. Accordingly, the most influential parameters, by a wide margin, are those directly determining the amount of storable

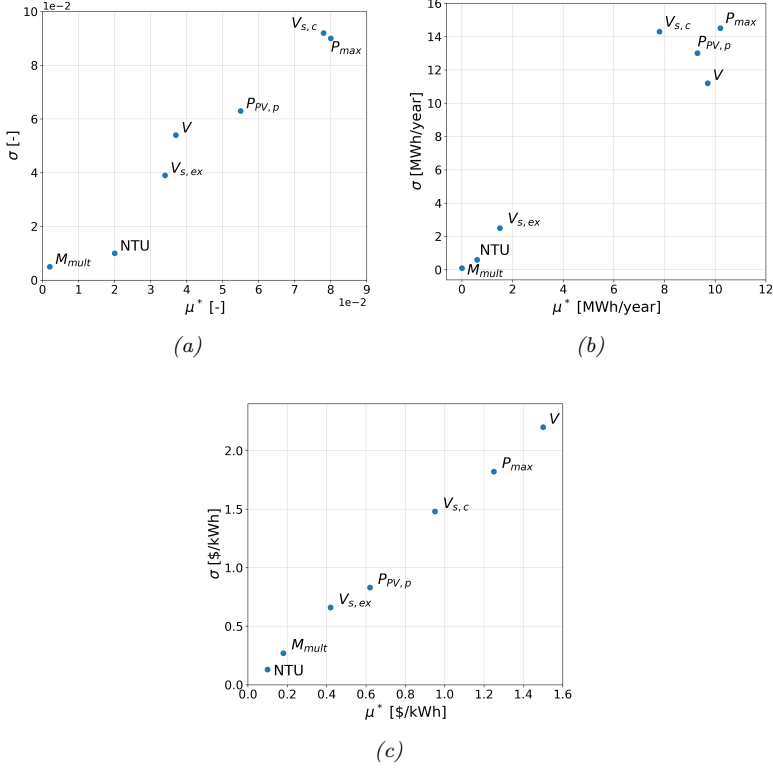


Figure 4.11: Morris sensitivity analysis results for: (a) RTE, (b)  $E_{out}$ , (c) LCOS.

energy, such as the maximum storage pressure and the reservoir volume, as well as the installed power and the compressor swept volume. The sizing of the photovoltaic system also plays a relevant role, since sufficient installed power is required to accumulate enough energy to compensate for supply shortages. The air storage volume and maximum pressure are equally critical, as their proper selection ensures that the system capacity is consistent with storage requirements. In addition, the compressor displacement significantly affects energy coverage, since, beyond its impact

on machine efficiency, it also determines the maximum instantaneous power that can be absorbed during the charging phase. Parameters related to heat recovery have a marginal impact on the system energy performance, both in terms of RTE and  $E_{out}$ . The oil mass multiplier ( $M_{mult}$ ) consistently shows a  $\mu^*$  value at least one order of magnitude lower than that of the other parameters. The NTU parameter is the second least influential: while its impact on RTE is of comparable magnitude, due to its effect on thermal recovery efficiency, its contribution to the output energy is significantly smaller. This behavior indicates that only a limited fraction of the recovered energy is associated with thermal storage, whereas the majority is stored in the form of compressed air.

Regarding the levelized cost of storage, the influence of the various parameters reflects a combination of their impact on capital expenditure (CAPEX) and their effect on the system capacity. As a result, the most influential parameters are those related to the overall system capacity, which must be carefully optimized to balance the increase in investment cost against the corresponding increase in electrical energy production. In this case, the oil mass multiplier exhibits a slightly higher influence compared to the other performance indicators, as its associated  $\mu^*$  is of the same order of magnitude as that of most other variables. This suggests that increasing the thermal oil mass would have a non-negligible effect on the system cost, even though its impact on efficiency remains limited.

Finally, it is worth noting that the comparable magnitudes of  $\mu^*$  and  $\sigma$  observed for most parameters indicate the presence of significant interaction effects among the input variables. This further confirms the importance of a holistic system design approach, in which the sizing of each component accounts for its mutual interactions with the rest of the system.

Based on the reported results, the oil mass multiplier and the design NTU are excluded from the set of parameters considered in the optimization process. Variations in these two parameters were found to have a negligible impact on both round-trip efficiency and energy output. There-



fore, it is reasonable to fix their values in order to limit the associated investment costs without significantly affecting the annual energy production of the system, ultimately yielding a benefit in terms of LCOS. With regard to the heat exchanger design, an NTU value of 4 under nominal operating conditions is selected. For this value, a counterflow heat exchanger operates close to the asymptotic limit of thermal effectiveness, as shown in Figure 4.12, ensuring high heat transfer efficiency while avoiding unnecessary increases in heat exchanger size and cost.

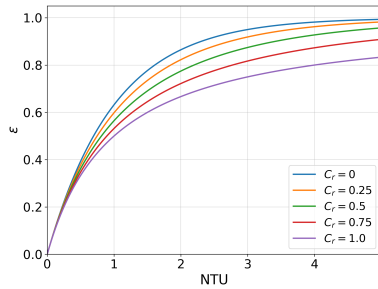


Figure 4.12: Efficiency of a counter flow heat exchanger

The oil mass multiplier was set equal to 1, as preliminary parametric analyses indicated that this is the minimum value that guarantees temperatures above 0  $^{\circ}\text{C}$  at the expander stages outlet. This choice prevents issues related to freezing during the expansion process. Although further reductions in this parameter could provide additional economic benefits, it would introduce significant technological constraints.

#### 4.5.2 Results of the optimization process

The results obtained from the optimization process are summarized in Table 4.6, both in terms of the parameter values that ensure the optimal performance with respect to the objective functions and in terms of the corresponding system performance.

The optimized RTE reaches a value of 40.1%. Although this value is relatively modest, it is fairly typical for simple CAES systems. Higher

Table 4.6: Optimal configurations predicted by EGO

| Objective function | $V_{s,c}$<br>( $\text{cm}^3$ ) | $V_{s,ex}$<br>( $\text{cm}^3$ ) | $V$<br>( $\text{m}^3$ ) | $P_{PV,p}$<br>( $\text{kW}$ ) | $p_{max}$<br>( $\text{bar}$ ) | $RTE$<br>(%) | $E_{out}$<br>( $\frac{\text{MWh}}{\text{year}}$ ) | $LCOS$<br>( $\$/\text{kWh}$ ) |
|--------------------|--------------------------------|---------------------------------|-------------------------|-------------------------------|-------------------------------|--------------|---------------------------------------------------|-------------------------------|
| RTE                | 3513                           | 16                              | 31                      | 84                            | 63                            | 40.1         | 29.9                                              | 0.734                         |
| $E_{out}$          | 4000                           | 21                              | 20                      | 176                           | 128                           | 30.5         | 36.4                                              | 0.908                         |
| LCOS               | 1598                           | 14                              | 12.1                    | 89                            | 127                           | 33.9         | 29.7                                              | 0.602                         |

efficiencies are generally achieved only at the expense of non-conventional components (e.g., isothermal compressors/expanders or constant-pressure storage systems) or more complex system layouts (e.g., a higher number of compression/expansion stages) [84, 200]. With regard to the economic optimization, the minimum achievable levelized cost of storage (LCOS) is 0.602  $\$/\text{kWh}$ . This value is higher than those reported for similar systems in the literature, which typically lie in the range 0.1–0.2  $\$/\text{kWh}$  [145, 201, 202]. The higher LCOS is mainly attributable to the limited efficiency of the system and to its relatively small scale. Increasing the system size, both in terms of rated power and installed energy capacity, would likely lead to improved economic performance, as also highlighted in the literature [134, 145].

Several considerations can also be drawn from the values of the optimized design parameters. As expected, high values of the maximum storage pressure and compressor swept volume are required to achieve high energy production. The compressor size reaches the upper limit of the investigated range, since this allows the system to absorb the maximum possible amount of energy from the photovoltaic source. Contrary to what might be intuitively expected, the installed PV capacity does not reach its maximum allowable value. Further increasing the PV size would mainly increase the fraction of the load directly supplied by the PV system, rather than the fraction effectively covered by the CAES.

In the configuration that maximizes the RTE, both the maximum storage pressure  $P_{\max}$  and the installed PV capacity ( $\text{kW}_p$ ) are significantly reduced. This configuration limits the amount of energy stored at pressures exceeding 45 bar, which corresponds to the maximum allowable inlet

pressure of the expander, and therefore reduces energy losses associated with throttling. Nevertheless, the optimal value of  $P_{\max}$  remains slightly above 45 bar. Storing air at marginally higher pressures can indeed be beneficial, representing a trade-off between minimizing throttling losses and allowing the expander to operate at higher pressure ratios, where its isentropic efficiency is maximized.

The configuration that minimizes the LCOS exhibits a substantial reduction in both compressor size and storage volume, while the maximum pressure remains comparable to that of the configuration maximizing energy production (approximately 120 bar). The cost breakdown of the three optimized configurations is reported in Figure 4.13. It can be observed that the costs associated with the compressor, the PV system, and the storage vessel are the most significant contributors to the total system cost; therefore, limiting their size is essential for achieving the economic optimum. These costs increase markedly when moving from the configuration with the minimum LCOS to that with the highest energy coverage, with the maximum-efficiency configuration lying in between. Although these increases lead to enhanced system capacity and performance, they do not necessarily translate into an overall economic benefit.

To check the convergence of the optimization process, the evolution of the expected improvement is plotted on Figure 4.14, for sake of brevity henceforth results are analysed only for the RTE optimization process, similar results are obtained for energy output and LCOS. Figure 4.14 shows that EI starts from a value above 0.06 and it decreases rapidly to reach a value below the convergence limit in 52 iterations. The decrease of the EI is associated not only with a reduction in the probability of finding an improved configuration, but also with a more correct prediction of the objective function by the surrogate model. As explained in the previous section, the value of the EI is based on both the predicted value of and the uncertainty of this prediction. This aspect is illustrated by Figure 4.15 that shows the comparison between the value predicted by the surrogate model and the value calculated by the thermodynamic model for each

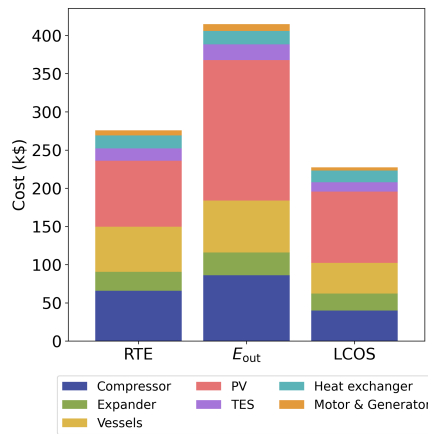


Figure 4.13: Investment cost breakdown

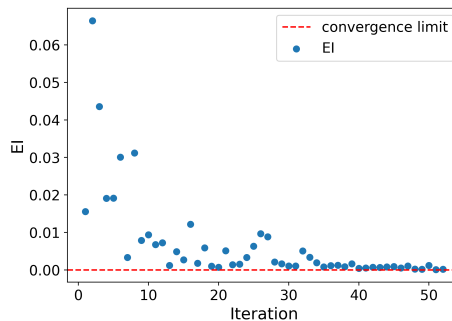


Figure 4.14: Evolution of expected improvement

iteration. The trend shows a clear decrease in the difference between the predicted value and the calculated value. These results show that the optimisation process has achieved adequate convergence and a reliable representation of the real model.

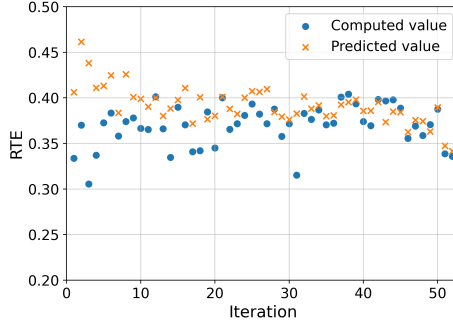


Figure 4.15: Comparison of predicted and exact value of RTE

## 4.6 Conclusion

This chapter introduces a revised micro-CAES configuration, incorporating modifications to the system layout derived from the conclusions of the previous chapters and from the current state of the art. The proposed architecture is based exclusively on commercially available and technologically mature components, ensuring feasibility at the micro-scale while preserving the generality of the modelling approach.

A comprehensive thermodynamic model has been developed to describe the behaviour of all system components under variable and off-design operating conditions. The model is employed to simulate the operation of the system in an off-grid configuration, coupled with a photovoltaic power source and domestic electrical loads. An operational strategy is implemented, whereby compression and expansion phases are driven by the instantaneous power imbalance between renewable generation and electrical demand, while respecting machine operating limits and storage constraints. This methodology avoids idealised cyclic assumptions and enables the evaluation of annual performance indicators under realistic operating profiles. From an assessment perspective, a consistent set of energetic and economic performance indicators is considered, including round-trip efficiency, delivered energy output, and the levelised cost of

storage. This evaluation framework allows the influence of component sizing on both system efficiency and long-term costs to be systematically investigated.

The developed model is subsequently employed within a sizing procedure, which represents the main focus of this chapter. This procedure consists of an initial sensitivity analysis followed by a dedicated optimisation phase. The thermodynamic model is coupled with an external optimisation platform (Dakota, Sandia National Laboratories) to perform both sensitivity analysis and optimisation. The sensitivity analysis, based on the Morris screening method, identifies the most influential design parameters affecting system performance. Less impactful parameters are therefore excluded from the subsequent optimisation stage, significantly reducing the computational burden and improving optimisation efficiency. In particular, the analysis highlights that the sizing of the thermal energy recovery subsystem has a comparatively limited impact on overall system performance. Consequently, a preliminary sizing approach can be adopted for this subsystem, primarily aimed at preventing freezing issues while simultaneously limiting capital costs. The optimisation phase, based on the Efficient Global Optimisation methodology, successfully identifies optimal system configurations targeting different objective functions (i.e. RTE, energy production, and LCOS). The use of an adaptively updated surrogate model allows convergence towards optimal solutions within a reasonable computational time, demonstrating the robustness and reliability of the proposed optimisation framework. The techno-economic analysis reveals clear limitations of the investigated configuration, which are mainly attributable to the intrinsic characteristics of micro-CAES systems. Nevertheless, it is important to emphasize that the primary objective of this chapter is not to assess the economic feasibility of the system, but rather to establish a reliable and flexible framework for subsequent evaluations.

Overall, this chapter presents a structured, robust, and computationally efficient methodology that, despite relying on a detailed thermodynamic model, enables the systematic assessment of sizing parameters

and the identification of optimal CAES designs. The proposed approach simultaneously accounts for component-level design and off-design performance. The following chapters build upon this modelling framework to analyse in greater detail the impact of operating conditions on both system-level and component-level performance, and to further investigate the techno-economic feasibility of micro-CAES systems in PV-coupled applications.

## Chapter 5

# Assessment of the Sizing Methodology through the Evaluation of Load Variability Effects

In this chapter, the modelling framework and sizing procedure previously developed are applied to assess their robustness and practical relevance under different operating conditions. In particular, the analysis investigates how the temporal variability of the electrical load influences both the optimal system design and the resulting energetic, exergetic, and economic performance. To this end, the system is evaluated under three representative demand profiles constant, moderately variable, and highly variable; allowing the impact of load dynamics on component optimal sizing, operational performance, and techno-economic feasibility to be systematically quantified. By applying the sizing and optimization procedure to each scenario, the analysis highlights how changes in operating conditions can lead to different optimal component sizes and performance levels. The results therefore highlights that integrated, scenario-aware modeling and optimization are required to avoid misleading conclusions





system performance under varying demand patterns.

- Scenario 1 is a constant load profile, representing a purely theoretical case used as a comparative reference to assess the impact of load variability.
- Scenario 2 corresponds to a statistical average of domestic electricity consumption [199]. It is representative of a CAES system supplying a sufficiently large number of small consumers whose unsynchronized demand smooths the overall load curve.
- Scenario 3 reflects the opposite situation, in which a limited number of larger consumers are connected to the CAES system, resulting in sharp and sudden variations in power demand. This load profile is constructed from the “TIER 3” reference demand described in [203].

All three load profiles are scaled so that the total daily energy consumption over a 24-hour period is equal to 200 kWh. The corresponding curves are shown in Figure 5.2. The variable demand scenarios (i.e. Scenarios 2 and 3) are selected based on the assumption that a realistic application of a small-scale CAES system would involve its integration into off-grid communities,

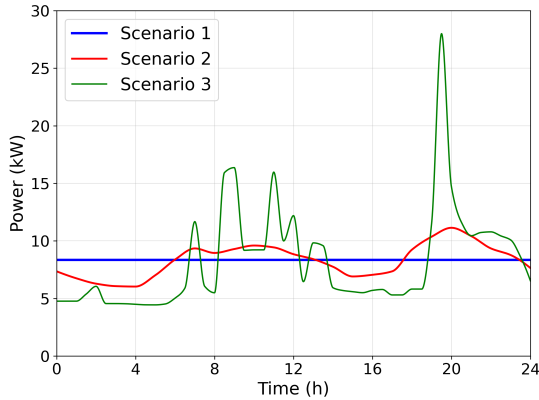


Figure 5.2: Load profiles

where electricity consumption typically exhibits morning and evening peaks. Although both scenarios reproduce domestic-like variability, they differ significantly in the magnitude and temporal distribution of the peak loads.

## 5.2 System modeling and performance metrics

The system is modelled from an energetic, exergetic, and economic perspective. The complete formulation is not reproduced here, as it has already been presented in the previous chapters. In particular, the energetic model is described in Sections 3.3 and 4.2, while the assumptions adopted for the exergetic formulation are reported in Section 3.4. Finally, the economic model and the estimation of investment costs are detailed in Section 4.3. The main performance parameters analysed in this chapter are summarised below. Some of them have already been introduced previously, whereas others are presented here for the first time.

- Energy performance is assessed by the Round-Trip Efficiency (Equation (3.22)), and the Renewable Self Sufficiency (RSS), which is defined as the ratio between the total energy released by the CAES system and the PV panel and the total energy request during the same period as shown by Equation (5.1). Both parameters are computed over a time span of a year.

$$RSS = \frac{E_{PV} + E_{ex}}{E_L} \quad (5.1)$$

- To assess the exergetic performance, two indicators are introduced: the Absolute Waste Exergy Ratio ( $\gamma_{A,i}$ ) and the Local Waste Exergy Ratio ( $\gamma_{L,i}$ ). These indicators, defined in Equation (5.2) and Equation (5.3), are computed for each component of the system. In these equations,  $\dot{E}x_D$  and  $\dot{E}x_L$  denote, respectively, the exergy destruction rate and the exergy loss rate to the environment. Both ratios quantify the fraction of exergy that is either destroyed or lost, normalised with respect to the total exergy input to the entire

system (for  $\gamma_{A,i}$ ) or to the individual component (for  $\gamma_{L,i}$ ). All exergy quantities are evaluated as time-integrated exergy flows over one year, consistent with the methodology adopted for the computation of the round-trip efficiency. The first indicator provides an estimate of the contribution of each component to the overall exergetic inefficiencies of the system. The second indicator, which may also be interpreted as the one's complement of the component exergetic efficiency, offers a direct measure of the intrinsic performance of each component, independent of its relative weight within the overall CAES system, thus enabling meaningful comparisons across different operating scenarios. No global exergetic efficiency is introduced for the entire system, since its only inputs and outputs are mechanical work; therefore, its exergetic efficiency coincides with the round-trip efficiency previously defined.

$$\gamma_{A,i} = \frac{\int \dot{E}x_D dt + \int \dot{E}x_L dt}{\int \dot{E}x_{in,s} dt} \quad (5.2)$$

$$\gamma_{L,i} = \frac{\int \dot{E}x_D dt + \int \dot{E}x_L dt}{\int \dot{E}x_{in,i} dt} \quad (5.3)$$

- The economic analysis is based on the evaluation of the Levelized Cost of Storage, previously defined in Equation (4.30). It shall be specified that the analysed scenario involves evaluating the cost obtained by considering the CAES system associated with a pre-existing photovoltaic system, resulting in a charging cost equal to zero. This approach is equivalent to assessing the feasibility of integrating a storage system into an off-grid configuration where a renewable energy generation system is already in place but lacks an associated storage system.

### 5.3 Performed optimization analysis

In each of the three scenarios analysed, both a single-objective and a multi-objective optimization are performed. The aim is to identify the

best possible configuration for each scenario, ensuring that the optimal solution is always selected within the explored design space.

- For the single-objective case, the selected objective function is the LCOS. The procedure relies on the Efficient Global Optimization algorithm [194], which is described in detail in Section 4.4.3.
- The multi-objective framework considers two objective functions, LCOS and RSS, with the aim of identifying the Pareto front within the design domain. Building on the results of the single-objective phase, a surrogate model is constructed to map each configuration to the corresponding objective values. The optimization is performed using a surrogate-based global strategy, in which a Gaussian Process surrogate is iteratively refined. At each iteration, a Multi-Objective Genetic Algorithm (MOGA) is applied to the surrogate to approximate the Pareto front. Only a limited subset of representative Pareto-optimal configurations is subsequently selected for evaluation using the thermodynamic model. The points are selected in order to promote both coverage of the Pareto front and refinement of sparsely sampled regions of the objective space. The resulting data points are added to the training set, progressively improving surrogate accuracy in the most relevant regions. Convergence is assessed a posteriori by monitoring the stabilization of the Pareto front across successive surrogate refinement iterations. A minimum of 20 surrogate refinement iterations is adopted as a baseline and increased if necessary. The main parameters controlling the genetic algorithm population evolution are reported in Table 5.1, while the remaining settings follow the default Dakota implementation.

The parameters varied during the optimization and their corresponding bounds are summarized in Table 5.2.

Table 5.1: Main genetic algorithm parameters used in the multi-objective optimization.

| Parameter                 | Value                 |
|---------------------------|-----------------------|
| Population size           | 300                   |
| Crossover operator (rate) | shuffle_random (0.8)  |
| Mutation operator (rate)  | replace_uniform (0.1) |

Table 5.2: Decision variables and bounds for the sizing optimization in the load-variability case study

| Parameter                       | Symbol     | Range                        |
|---------------------------------|------------|------------------------------|
| Compressor swept volume         | $V_{s,cc}$ | $500 \div 4000 \text{ cm}^3$ |
| Expander swept volume           | $V_{s,ex}$ | $10 \div 60 \text{ cm}^3$    |
| Pressure vessels volume         | $V$        | $10 \div 50 \text{ m}^3$     |
| Max air pressure inside vessels | $p_{max}$  | $40 \div 200 \text{ bar}$    |

## 5.4 Results and discussion

Results of the single objective optimization process, in terms of sizing parameters and system performance indices, are reported in Table 5.3. Results show significant similarities in terms of design parameters, particularly if compared to the range in which they are allowed to vary during the optimization; nevertheless different optimal designs are identified for the three scenarios, especially for storage volume and maximum pressure of the high variability scenario. Similar component sizing results in comparable investment costs, as shown in Figure 5.3 presenting the cost breakdown of the three optimal configurations. The bar plots illustrates how air vessel and compressor are the components with the higher cost.

Regarding system performance, limited variations are observed in terms of RTE which varies from a maximum of 34.1% to a minimum of 33.7%. Renewable self sufficiency shows comparable values, it is equal to 89.4% , 90.5% and 87.6% for the three scenarios, showing a maximum for Scenario 2 and a minimum for Scenario 3. Figure 5.4 presents the comparison between the required and the produced power for the three different configurations.

Table 5.3: Optimal configurations to minimize LCOS

| Scenario | $V_{s,c}$<br>( $cm^3$ ) | $V_{s,ex}$<br>( $cm^3$ ) | $V$<br>( $m^3$ ) | $p_{max}$<br>(bar) | RTE<br>(%) | RSS<br>(%) | LCOS<br>(\$/kWh) |
|----------|-------------------------|--------------------------|------------------|--------------------|------------|------------|------------------|
| 1        | 1274                    | 13                       | 12.5             | 121                | 33.7       | 89.4       | 0.366            |
| 2        | 1333                    | 14                       | 12.4             | 120                | 33.3       | 90.5       | 0.371            |
| 3        | 1355                    | 15                       | 15.4             | 94                 | 34.1       | 87.6       | 0.393            |

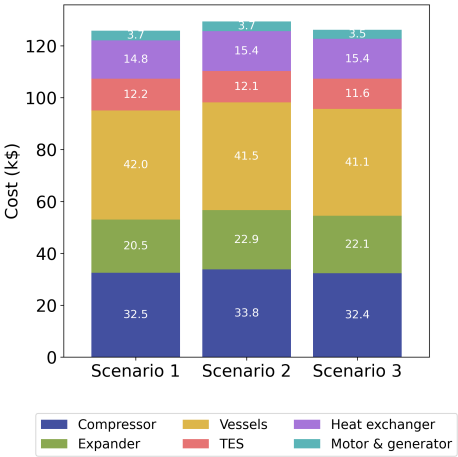


Figure 5.3: Cost breakdown for configurations that minimize LCOS

The graphs show that, in the first two scenarios, the system can deliver the maximum power required by the load during each month. While in the summer it successfully covers the entire interval not supplied by solar energy, in the other months a portion of the demand remains unmet by the CAES system. This limitation arises from the system’s inability to store sufficient energy during the daytime. A different situation is observed for the third scenario, where the CAES system can never deliver the total required power at peak demand times. This limitation is not due only to the system’s energy storage capacity but to the combination of storage pressure and expander size, which is insufficient to meet the peak power

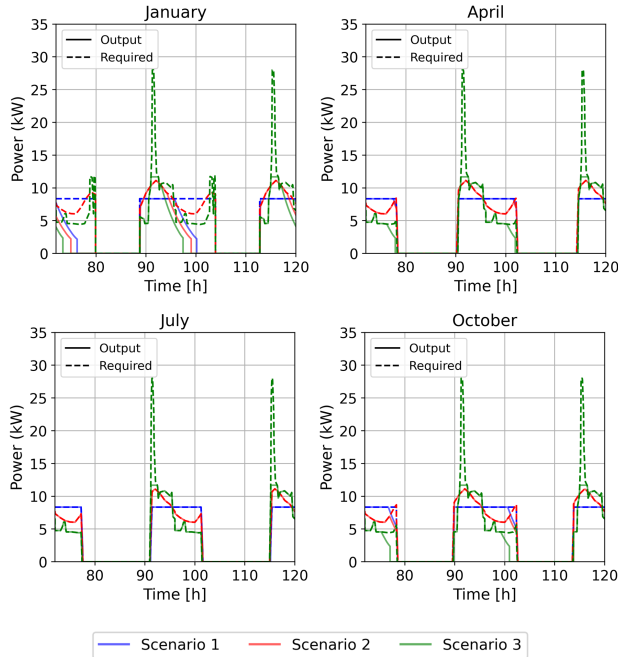


Figure 5.4: Comparison of power request and power output for optimized configurations

requirement. As will be detailed in the following sections, the algorithm tends to select an undersized expander relative to the peak load because covering this peak would result in an increased cost per kWh, despite the additional energy provided to the system. The images presented here illustrate the trends of the variables over a 48-hour time span, ranging from 72 to 120 hours within the simulated interval. This visualization approach was chosen to better highlight the differences between the various cases. As previously mentioned, the simulations were conducted over a one-week period for each month. The results exhibit a brief transient phase before stabilizing into a cyclic daily pattern. By displaying the data in this manner, the analysis focuses on the period following the



transient phase, showing a complete charge and discharge cycle of the system, which is representative of the corresponding month. For brevity, in the following sections, results will be presented for a single month, as the main differences between the three configurations remain consistent regardless of the analyzed season. Finally, the computed leveled cost of storage is respectively  $0.366\$/kWh$ ,  $0.371\$/kWh$  and  $0.393\$/kWh$  for the three scenarios. Such values are higher than those usually reported for CAES systems, which are in the range of  $0.1\text{--}0.15\text{ } \$/kWh$  [145], but mainly refer to larger-scale systems. Values reported in this work are limited by both system efficiency and scale. It is also worth noting that, as specified in Section 5.2, the values computed in this chapter assume a pre-existing photovoltaic system, and therefore a zero charging cost. This explains the discrepancy between the LCOS values reported here and those presented in the previous chapter, where the installation cost of the PV system was included in the economic assessment.

To better illustrate the impact of the load, the performance of the three configurations obtained during the optimization phase for each load is also evaluated under the conditions of the other two loads. The results are presented in Table 5.4. By moving along a row in the table, the results correspond to the same system sizing varying the associated load. Conversely, by moving along a column, the load remains constant while the system sizing changes from one optimal configuration to another, as determined in the previous phase. The results along the diagonal of the matrix represent the optimal configurations previously presented. By examining the results reported in the table, it emerges that, for each of the three sizing configurations, Scenario 1 and Scenario 2 yield comparable cost performances, whereas a noticeable deterioration occurs when moving to Scenario 3. The most pronounced increase in LCOS is observed for Sizing 1, where the metric rises by  $0.035\$/kWh$  when transitioning from Scenario 1 to Scenario 3. When moving from Scenario 1 to Scenario 2, the LCOS may increase even in the presence of a slight improvement in RSS, as seen for Sizing 2. This occurs because, by definition, the renewable self-sufficiency index also includes the fraction of the load directly supplied

Table 5.4: Optimal configurations performance under different loads request

| Sizing | Scenario 1 |         |                      |               | Scenario 2 |         |                      |               | Scenario 3 |         |                      |               |
|--------|------------|---------|----------------------|---------------|------------|---------|----------------------|---------------|------------|---------|----------------------|---------------|
|        | RTE (%)    | RSS (%) | $E_{out}$ (MWh/year) | LCOS (\$/kWh) | RTE (%)    | RSS (%) | $E_{out}$ (MWh/year) | LCOS (\$/kWh) | RTE (%)    | RSS (%) | $E_{out}$ (MWh/year) | LCOS (\$/kWh) |
| 1      | 33.72      | 89.45   | 30.31                | 0.366         | 32.65      | 89.07   | 29.44                | 0.377         | 31.51      | 86.66   | 27.71                | 0.401         |
| 2      | 33.89      | 90.23   | 30.88                | 0.367         | 33.34      | 90.51   | 30.50                | 0.371         | 32.32      | 88.45   | 29.02                | 0.390         |
| 3      | 34.81      | 88.20   | 29.39                | 0.380         | 34.78      | 89.04   | 29.42                | 0.379         | 33.97      | 87.58   | 28.38                | 0.393         |

by the photovoltaic system. Although the three scenarios share the same total daily energy demand, their distinct temporal profiles lead to different interactions with the solar resource. As a consequence, the amount of energy that must be delivered by the CAES system varies slightly, as highlighted in the table, which also reports the total annual energy output of the CAES unit over its operational lifetime ( $E_{out}$ ). In this case, the increase in RSS does not correspond to a higher CAES energy output. On the contrary,  $E_{out}$  is lower in Scenario 2, resulting in an increased LCOS. This indicates that the marginally higher renewable coverage is attributable to a greater contribution from the photovoltaic system, due to a more favorable temporal alignment between PV generation and the load profile of Scenario 2. Regarding RTE, it always reaches the lowest value for Scenario 3, with an average decrease of approximately 1.5 percentage points compared to the best one. Moving across each column, the observed variations are less significant, with the largest LCOS difference reaching around 0.013 when deviating from the optimal sizing. The highest efficiencies are always observed in Sizing 3, primarily due to the reduction in storage pressure, which minimizes throttling losses. Sizing 3 is characterized by a lower storage pressure to better match with the third load scenario, as it will be detailed in Section 5.4.2.

In terms of design guidelines, results here reported show that assuming a simplified constant load during the design phase leads to equipment dimensions very similar to those derived from a profile-specific optimization, supporting its use as a practical sizing strategy (i.e. limited variation moving along the rows of the table). Nevertheless, it remains essential to test the system under actual, time-varying loads, since its real-world

performance can significantly diverge from the case of constant load (i.e. higher variations moving along the column). In the following sections, a more detailed analysis will be conducted to understand how the load affects individual components performance and how these effects propagate, influencing the overall system performance. Particularly the results will be shown considering the case in which the optimal sizing for the constant load is operated with the three considered loads and the case in which the three optimal sizing are operated in combination with the load exhibiting maximum variability. This corresponds to comparing the results reported in the first row and the last column of Table 5.4, respectively.

#### 5.4.1 Performance comparison for a fixed sizing under different loads condition

This section presents the simulation results for the system sized according to the optimization performed under the constant load scenario (hereafter "Sizing 1"), tested against all three load demands. This approach allows for the analysis of the variations that a system, preliminary optimized for an constant load, would experience when it operates under real conditions defined by a specific load profile. Since the same design is considered (i.e. same components cost) considerations taken regarding the energy/exergy aspects directly reflect to LCOS considerations. Results obtained from the exergy analysis in the three different scenarios are shown in Figure 5.5 and Figure 5.6. The figures also report the results obtained by simulating the same system sizing under conditions of ideal operability, by performing a sequence of full charge and discharge cycles while operating the compressor and expander at the rotational speeds that yield maximum efficiency for each pressure ratio, without the constraint of the presence of any external load. This approach allows to quantify the deviation of off-design operation from the maximum efficiency achievable by the system design.

From the chart in Figure 5.5, it is clear that the highest exergy destruction and losses occur in the compressor and expanders, accounting for approximately 35% and 20% of the system's input energy, respectively.

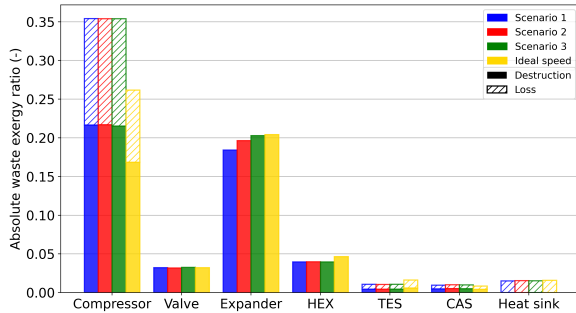


Figure 5.5: Absolute waste exergy ratio under different load scenarios

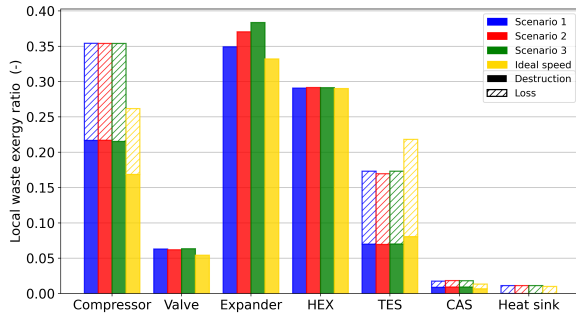


Figure 5.6: Local waste exergy ratio under different load scenarios

All other components contributes are one order of magnitude lower. By comparing the histograms for the three scenarios, it can be observed that while the compression phase shows almost no variation, significant differences emerge during the expansion phase. In particular, the exergy destruction associated with the expander increases from 18.4% to 20.3% in absolute terms (Figure 5.5), and from 34.9% to 38.3% in relative terms (Figure 5.6). Only minimal differences are observed for the remaining components. This finding also helps explain the limited variation in round-trip efficiency among the three scenarios, as the majority of exergy losses are concentrated in the compressor. The reasons behind these

results will be explored in greater detail in the following sections, where the time evolution of the operating conditions of each component will be analyzed. By comparing the three scenarios with the ideal case, whose round-trip efficiency reaches a value of 40.5%, a significant reduction in exergy wasted during the compression can be immediately observed, amounting to approximately 9%. Regarding the expansion phase, although the absolute exergy destruction in the ideal case is higher than in the other three scenarios, this is primarily due to the fact that a larger amount of exergy reaches the inlet of the expander. In fact, when considering the relative destruction, the ideal case obviously shows a lower value. The other system components do not show significant variations, except for the exergy loss associated with the TES. This is attributed to the longer duration of the compression and expansion phases for the ideal scenario, as the machines process a reduced mass flow rate. Consequently, the system tends to dissipate a larger portion of energy to the external environment. Again, its overall contribution is still very limited

#### **5.4.1.1 Impact on compressor performance**

As shown in Figure 5.6 the impact of load variability on compressor efficiency is negligible. The reason of this is summarized in the temporal graphs reported in Figure 5.7. As shown, the available power, defined as difference between the power generated by the photovoltaic system and the power consumed by the load, is minimally affected by load variations, as the former far exceeds the demand. The compressor quickly reaches the maximum rotational speed, leading to a significant amount of curtailment. Therefore, the compressor operates under very similar conditions in all three cases (Figure 5.7-b), resulting in the same level of efficiency (Figure 5.7-c).

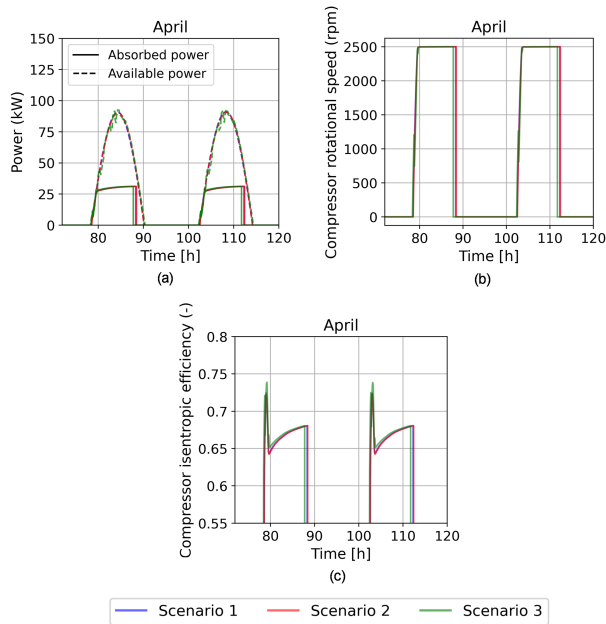


Figure 5.7: Compressor operating conditions under different load scenarios

#### 5.4.1.2 Impact on expander performance

Moving onto the analysis of the expansion phase, as illustrated in Figure 5.8, the system primarily responds to the variation of load and inlet pressure by adjusting its rotational speed to modulate the mass flow rate. As long as the inlet pressure is stable at 45bar (i.e. stored pressure higher than maximum expander operating pressure, and throttling occurring in the valve), the rotational speed vary according to the power; when the pressure decreases the rpm rapidly raises to increase the mass flow rate and balance the reduction of specific work due to lower pressure ratio. This adjustment is the only control method for the system under consideration. Changes in rotational speed influence not only the mass flow rate but also the expansion efficiency (Figure 5.8-e). Figure 5.8-

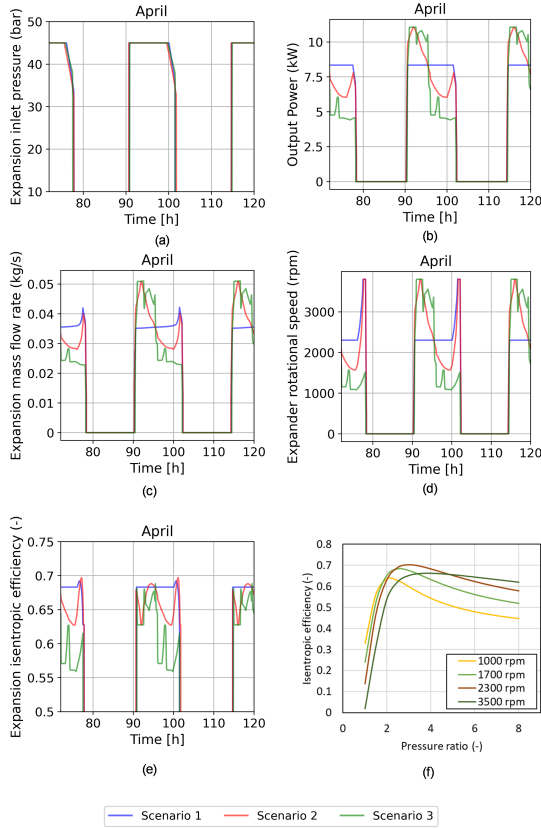


Figure 5.8: Expander operating conditions under different load scenarios

f shows the isentropic efficiency of a single expander as a function of the expansion ratio and rotational speed. The expander size for sizing 1 is selected in such a way that, in Scenario 1, the expander operates for most of the time at a rotational speed close to the optimal. Other configurations, operating at higher or lower rotational speeds, exhibit lower efficiencies. This reduction in efficiency directly impacts the global RTE, which progressively decreases from the constant load scenario to

the third scenario.

### 5.4.1.3 Impact on heat recovery system

The variation of air mass flow rate also affects the heat recovery system, primarily by altering the global heat transfer coefficient of the heat exchangers, as shown in Figure 5.9-a (the quantities shown in the graph presented in Figure 5.9 refer to the heat exchanger located before the inlet to the first expansion stage).

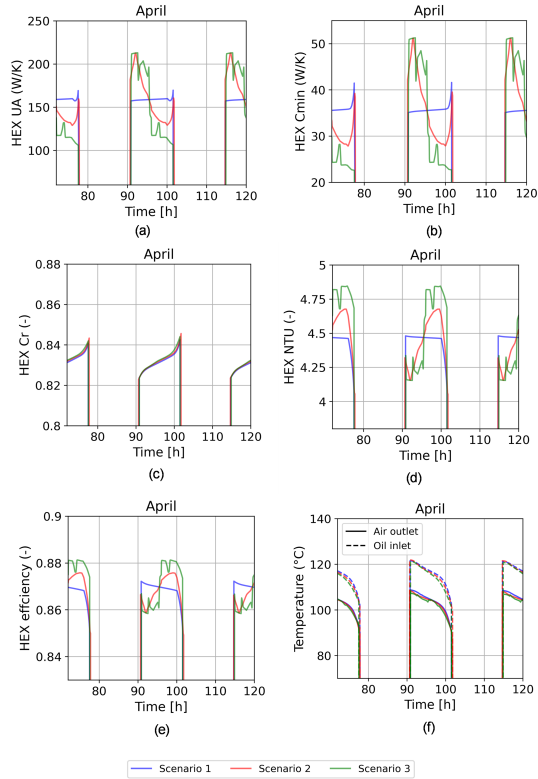


Figure 5.9: Heat exchanger performance under different load scenarios



However, this change in UA value does not significantly impact the efficiency of the heat exchanger(Figure 5.9-e). As introduced in Section 4.2.4 the system implements a control to regulate the mass flow rate of oil to be equal to 0.55 times the air mass flow rate, leading to an approximately constant value of the thermal capacity ratio ( $Cr$ ). As shown in Figure 5.9-d the value of the number of transfer unit ( $NTU=UA/C_{min}$ ) remains between 4 and 5: this, combined with the almost constant value of  $Cr$ , leads to a nearly constant efficiency, as expected from the theoretical description of  $\varepsilon$ -NTU method [204]. The expander inlet temperature (i.e. air outlet temperature from the heat exchanger) is primarily influenced by evolution of the outlet temperature of the diathermic oil from the thermal storage rather than by the HEX efficiency as shown in Figure 5.9-f. The reported graph highlights how the inlet oil temperature to the heat exchanger remains mostly unchanged varying the load type. A slightly greater, but still negligible, difference is observed for the air temperature, which is influenced by both the inlet temperature and the efficiency of the heat exchanger. The temperature difference between the two streams of the HEX shown, justifies the non-negligible exergy destruction that occurs in the heat exchangers, as depicted in the graph in Figure 5.6.

#### 5.4.2 Performance comparison of multiple sizings coupled with fixed load condition

The following section analyzes the system's behavior, sized according to the three configurations, in "Scenario 3". This provides a deeper understanding of how performing an optimization tailored to the specific case leads to only limited improvements compared to considering a design optimized for a stationary load, and highlights the differences observed between different sizing. Despite comparing different designs and consequently different costs, their variation is limited (Figure 5.3); therefore components inefficiencies still constitutes also the primary cause of LCOS variations.

Starting the comparison from the results of exergy analysis reported in Figure 5.10 and Figure 5.11, it is evident that the increase in round-

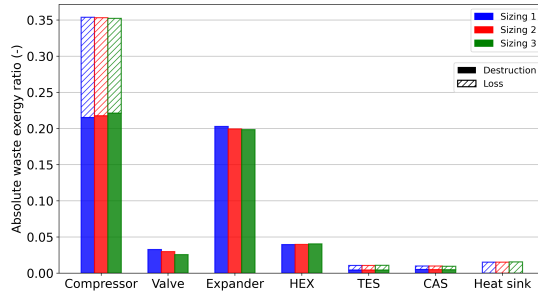


Figure 5.10: Absolute waste exergy ratio of different sizing coupled with fixed load condition

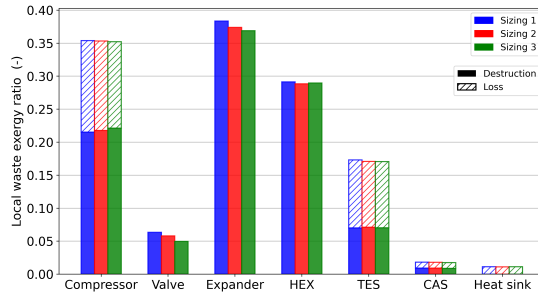


Figure 5.11: Local waste exergy ratio of different sizing coupled with fixed load condition

trip efficiency from sizing 1 to sizing 3 is due to a reduction in exergy destruction within the control valve and the expander. While in the case of the valve this is primarily due to the reduction in storage pressure, for the expander it is the result of a combination of effects.

Figure 5.12-b shows that Sizing 3 configuration tends to operate at rotational speed closer to its optimal operating point ( $2000 \div 2200$ ), thus achieving slightly higher efficiency (Figure 5.12-c). It is worthily to notice that this variations of efficiency are limited if compared to the ones previously shown in Figure 5.8-e where a specific system design operates under different load conditions. The variation in rotational speed

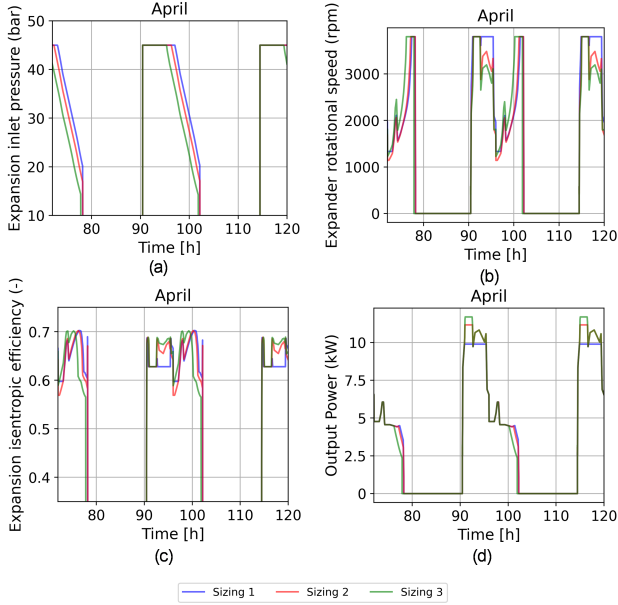


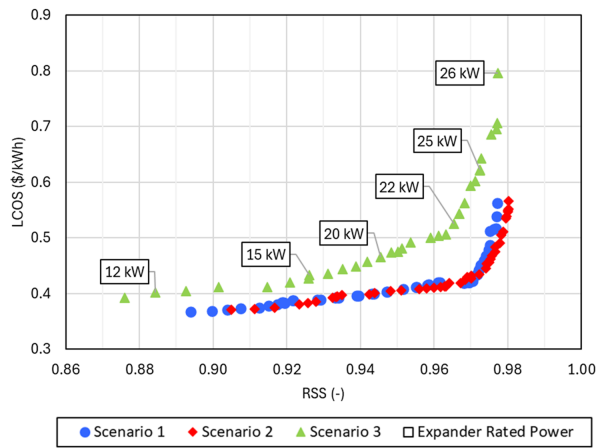
Figure 5.12: Expander operating condition with different sizing

is attributed to both the different expander size and the storage pressure. In sizing 3, the reduced storage pressure leads to a lower expansion inlet pressure for roughly half of the expansion phase, as illustrated in Figure 5.12-a. This enables operation at slightly higher efficiency during the interval between approximately 95 and 100 hours, as shown in the figure.

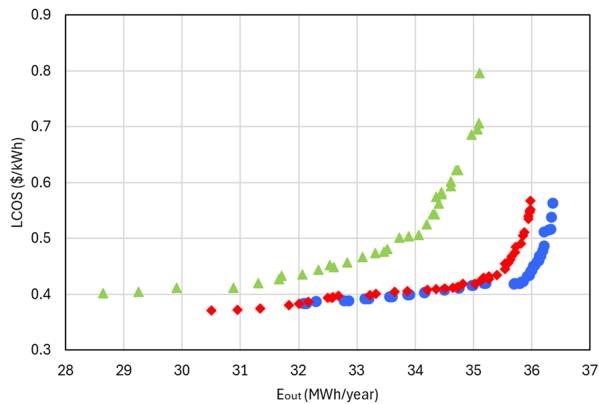
### 5.4.3 Results of multi objective optimization

It is also of interest to evaluate the modifications required to enhance the self sufficiency in different scenarios: considering off-grid applications, a design requirement could be to achieve renewable coverage close to 100%. For this purpose, Pareto frontiers are constructed in the RSS – LCOS plane and shown in Figure 5.13-a. The front corresponding to the first two

configurations are almost overlapping. In these configurations, the LCOS



(a)



(b)

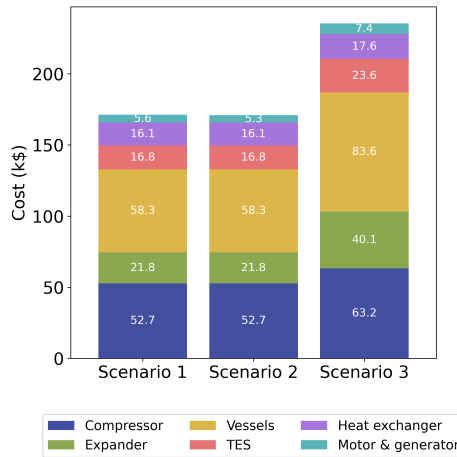
Figure 5.13: RSS-LCOS Pareto front: (a) plotted as a function of the renewable self-sufficiency; (b) plotted as a function of the storage energy output

slowly raises as the RSS increases up to approximately 97.5%; beyond this point, the cost rises exponentially. In these configurations, increasing RSS primarily requires increasing the compressor size to absorb more energy from the photovoltaic system to address the unmet load. Initially, the cost associated with the larger compressor size is partially counterbalanced by the higher energy output, so that the LCOS increases slowly. For Scenario 3, the overall trend is similar but the cost increases more rapidly with the required energy. In this case, the increase in compressor size is compounded by the need to cover peak power demand, resulting in a larger installed expander capacity, as visible from the note reported in the scatter plot. This further increases the investment cost without substantially enhancing renewable self sufficiency, as only a limited amount of energy is supplied during peak periods, while the demand during off-peak times is already met by a smaller expander. Ultimately, this leads to a more rapid rise in the levelized cost of storage. Figure 5.13-b shows the Pareto front points plotted as a function of the energy delivered by the CAES system. In this case, a difference between Scenario 1 and Scenario 2 can be observed, with the latter exhibiting a higher cost for the same amount of energy delivered. The difference between these two scenarios becomes evident only through this representation, as Scenario 2, characterized by a peak during midday hours, presents a better matching with photovoltaic energy availability for the same total daily energy demand, as already noticed in Section 5.4 when illustrating the performance of the configurations that minimize LCOS. As a result, Scenario 2 ensures the same RSS percentage as Scenario 1 with a lower amount of energy supplied by the CAES system. The higher cost associated with the same output energy in Scenario 2 is attributable to the greater investment cost of the expander, which is required to deliver a higher power output.

To further analyze what happens as RSS increases, a specific coverage value is fixed and the corresponding configuration on the Pareto frontier is analyzed. Table 5.5 illustrates the configuration that achieves a value of RSS equal to 97%. A marked difference in both sizing parameter and performance is now observed between the first two scenarios and the third.

Table 5.5: Configurations on the LCOE-RSS Pareto front at  $RSS=97\%$ 

| Scenario | $V_{s,c}$<br>( $cm^3$ ) | $V_{s,ex}$<br>( $cm^3$ ) | $V$<br>( $m^3$ ) | $p_{max}$<br>(bar) | RTE<br>(%) | RSS<br>(%) | LCOS<br>(\$/kWh) |
|----------|-------------------------|--------------------------|------------------|--------------------|------------|------------|------------------|
| 1        | 2536                    | 12                       | 17.2             | 120                | 33.8       | 97         | 0.424            |
| 2        | 2326                    | 14                       | 19.1             | 109                | 33.4       | 97         | 0.427            |
| 3        | 3029                    | 33                       | 32.1             | 91                 | 28.9       | 97         | 0.594            |

Figure 5.14: Cost breakdown for configurations that ensure  $RSS=97\%$  in different scenarios

The configuration of Scenario 3 shows a notable increase in the size of the expander, which is required to supply peak power. The compressor and storage volume are also significantly larger, partly due to the lower RTE of this configuration (28%) compared to the others (approximately 33%). As a result, a greater amount of energy must be stored to meet the same energy demand. These larger component sizes lead to a marked increase in the system's capital cost, as illustrated in Figure 5.14.

Regarding the RTE, while in the first two configurations, it does not deviate significantly from the value obtained in the minimum LCOS case,

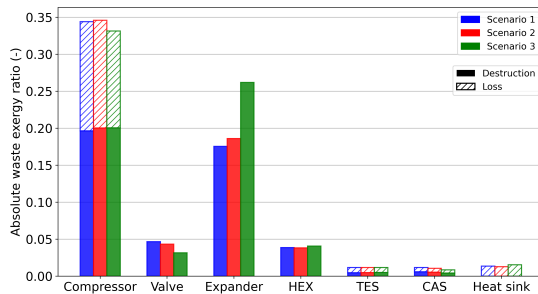


Figure 5.15: Absolute waste exergy ratio in configurations that ensure 97% of self sufficiency

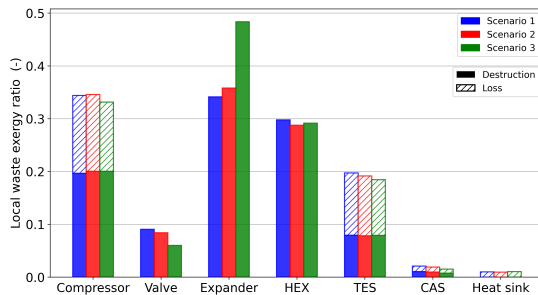


Figure 5.16: Local waste exergy ratio in configurations that ensure 97% of self sufficiency

for the high variability load it drops to 28.9%. This drastic reduction is primarily attributable to the expanders, as shown by the results of the exergy analysis in Figures 5.15 and 5.16. Their exergy destruction increases from approximately 18% of the input energy to over 25%, as they consistently operate at very low rotational speed and efficiency as shown in Figure 5.17. Obviously, LCOS variations are affected in this case not only by components inefficiencies but also by costs, as they significantly differ among the configurations: while the scenarios 1 and 2 exhibit an LCOS of \$0.424/kWh and \$0.427/kWh, respectively, the third scenario shows a significantly higher cost of \$0.594/kWh, due to the increased

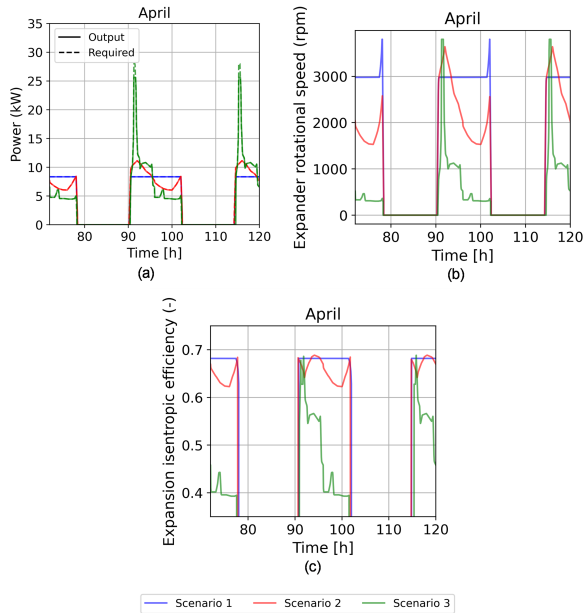


Figure 5.17: Expander operating conditions under different load scenarios and  $RSS=97\%$

components dimensions.

## 5.5 Conclusion

This chapter has served to assess the sizing and optimization methodology developed in the previous chapter, by testing its capability to capture variations in optimal design that lead to different system performance under different operating conditions. Moreover, it has presented the assessment of the influence of load variability on the performance, and cost of a micro-CAES system integrated with photovoltaic generation. The analysis was carried out using a dynamic model, combined with exergy analysis and techno-economic optimization, enabling a consistent evaluation of both component and system-level performance.



The main findings emerging from the work are the following:

- CAES performance is intrinsically influenced by load variability. Even when components are optimized for each specific scenario, increasing load variability leads to reduced energetic and economic performance. For the analyzed case study, moving from a constant-load profile to a highly variable demand results in an increase of the LCOS by 0.027 \$/kWh and a reduction of the annual energy output by 1.93 MWh.
- From a design perspective, when minimizing the LCOS is the primary objective, assuming a constant load leads to system sizing very close to that obtained under realistic load profiles, supporting its use as a reasonable simplification for preliminary design. However, evaluating real load profiles remains essential to properly assess the energy, exergy, and economic performance of the system in a specific scenario. In this context, the application of the proposed sizing and optimization procedure enables a scenario-aware assessment and the identification of design solutions that achieve incremental improvements in system performance.
- Load variability significantly impacts system operation, particularly during the expansion phase, forcing the expander to operate away from its nominal point. For the analyzed case study, expander exergy destruction increases from approximately 18.4% to 20.3% when a system optimized under constant-load assumptions is operated under highly variable demand. This results in a reduction of round-trip efficiency and overall cost-effectiveness. Compressor, heat exchanger, and thermal storage performance remain comparatively robust, showing only limited sensitivity to load dynamics across the analyzed scenarios.
- Exergy losses are primarily concentrated in the compression and expansion processes, with the compression phase contributing the most under all operating conditions. Heat recovery and thermal storage systems play a secondary role, with a limited impact on

overall system efficiency.

- When high levels of renewable self-sufficiency are targeted, accurate load representation becomes critical even at the design stage, and the impact of the sizing and optimization procedure becomes even more evident. For a target RSS of 97%, substantial discrepancies in both optimal component sizing and system performance emerge between scenarios. Highly variable load profiles require significantly larger components, particularly expanders, leading to lower round-trip efficiencies (down to approximately 29%) and higher LCOS values (up to 0.59 \$/kWh for the considered case study).
- The minimum LCOS value obtained (0.366\$/kWh) is higher than those of large-scale CAES or battery systems. These values reflect both the limited efficiency and fixed cost burden typical of standard micro-CAES technologies.

Overall, the results indicate that micro-CAES systems are best suited for applications characterized by stable or moderately variable loads, or as part of hybrid configurations in which load smoothing strategies can be implemented. Beyond these system-level considerations, the chapter demonstrates the importance of adopting an integrated and scenario-aware sizing methodology to obtain reliable assessments of optimal system design and performance, particularly when additional constraints are introduced in the problem formulation. Accurately accounting for load variability during the early design phase is therefore necessary to avoid underestimating both performance losses and economic penalties. The following chapters therefore build upon this methodology to address the thesis objective.



## Chapter 6

# Environmental and Economic Viability of Hybrid Micro-CAES Systems for Remote Off-Grid Communities

This chapter represents the first closing point of the research objectives of the thesis. After the development and validation of the micro-CAES system model and the sizing and optimization methodology in the previous chapters, the focus is here shifted towards their integrated application to a realistic system-level case study. In this chapter, the micro-CAES system model and the sizing methodology previously developed and validated are used to evaluate the sustainability of hybrid energy configurations for remote off-grid communities. These areas typically rely on diesel generators, which involve high operating costs, fuel supply uncertainties, and considerable environmental burdens. The integration of renewable energy sources represents a promising alternative, but their variability requires complementary storage systems capable of ensuring reliability and continuity of supply. For this reason, combining photovoltaic generation with diesel backup and micro-CAES storage emerges as a particularly rel-

evant configuration to investigate. The analysis presented here integrates thermodynamic modelling, optimization procedures, and a Life Cycle Assessment framework based on parametric inventories, which enable the quantification of environmental impacts as a function of system size and configuration. In parallel, the economic assessment relies on the Levelized Cost of Energy. Three configurations—diesel-only, PV–diesel, and PV–diesel–CAES—are compared across multiple sites characterized by different solar resources and fuel price conditions. This comparative approach allows the identification of the conditions under which micro-CAES integration provides meaningful environmental benefits and whether such improvements can be achieved while maintaining competitive energy costs.

## 6.1 System description and analyzed scenarios

The analyzed system consists of an off-grid end user whose electricity demand must be continuously satisfied according to a prescribed load profile. Given the absence of any connection to the main electrical grid, the reliability of the local energy supply is ensured through an appropriate combination of generation, storage, and back-up technologies.

To quantify the environmental and economic benefits associated with system hybridisation, three operational scenarios are defined and compared against a diesel-only baseline:

- Scenario 1: the user load is supplied exclusively by the diesel generator;
- Scenario 2: the PV plant provides electricity whenever solar energy is available, while the diesel generator compensates for the residual demand;
- Scenario 3: the hybrid system integrates an adiabatic micro-CAES unit, which stores surplus PV electricity as compressed air and subsequently releases it during periods of low solar availability. In this configuration, the diesel generator functions strictly as a back-

up unit, intervening only when both PV generation and stored energy are insufficient to satisfy demand.

This system architecture builds upon the configurations introduced in the previous chapters, particularly regarding the configuration of the CAES unit. In this analysis, however, the mechanism for supplying unmet demand is explicitly considered: any portion of the load not covered by PV generation or storage discharge is assumed to be provided by a back-up diesel generator, ensuring continuous power availability.

Figure 6.1 schematically illustrates the system and the three scenarios.

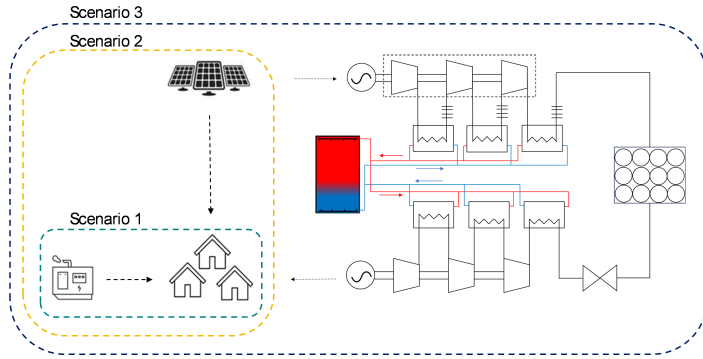


Figure 6.1: Schematic diagram of the diesel-PV-CAES system

## 6.2 Methodology

The system is analyzed from a techno-economic and environmental perspective. This section presents the modeling approaches adopted. With regard to the energy and economic analysis, the description focuses mainly on the diesel generator and its control logic, as this is the only element that differs from the configurations discussed in the previous chapters. Conversely, the environmental assessment is addressed in greater detail for all system components, since this is the first point in the thesis where the corresponding modeling framework is introduced.

### 6.2.1 Energy analysis

The power output of the diesel generator is given by

$$P_{\text{DG}} = \dot{m}_{\text{fuel}} \cdot LHV_{\text{fuel}} \cdot \eta_{\text{DG}}, \quad (6.1)$$

where  $\dot{m}_{\text{fuel}}$  and  $LHV_{\text{fuel}}$  denote, respectively, the fuel mass flow rate and the lower heating value of the fuel, while  $\eta_{\text{DG}}$  is the generator efficiency. The diesel generator is assumed to operate with a constant average efficiency equal to 0.35. The diesel generator, like all mechanical components, exhibits efficiency penalties under off-design operating conditions. However, preliminary analyses showed that adopting a constant efficiency assumption leads to negligible variations in the overall system performance indicators and does not alter the conclusions drawn from the analysis. Therefore, a constant average efficiency is assumed in this work in order to simplify the model.

The diesel generator supplies the amount of power required to satisfy the instantaneous power balance between the load demand and the power provided by the rest of the system. Depending on the scenario, this supply may consist solely of diesel power, diesel and photovoltaic power, or a combination of diesel, photovoltaic, and CAES power. Priority is given to renewable sources (i.e. photovoltaic generation and the CAES unit) so the diesel generator is dispatched only when these sources are insufficient to meet the load.

Accordingly, the diesel generator power is defined as follows:

- Scenario 1:

$$P_L(t) = P_{\text{DG}}(t). \quad (6.2)$$

- Scenario 2:

$$P_{\text{DG}}(t) = \max \{0, P_L(t) - P_{\text{PV}}(t)\}. \quad (6.3)$$

- Scenario 3:

$$P_{\text{DG}}(t) = \max \{0, P_L(t) - P_{\text{PV}}(t) - P_{\text{CAES}}(t)\}. \quad (6.4)$$

The CAES control strategy follows the same principles described in Section 4.2.5. The models of the remaining system components are described in Sections 3.3 and 4.2.

Energy performance is assessed by the al Round-Trip Efficiency and the Renewable Self Sufficiency, both parameters are computed over a time span of a year.

### 6.2.2 Economic analysis

The economic analysis is based on the evaluation of the global Levelized Cost of Energy of the entire system, which accounts for the total energy supplied by all generation sources, as defined in Equation (6.6). The investment cost of each component is estimated using the corresponding cost functions and subsequently updated using the Chemical Engineering Plant Cost Index (CEPCI). The capital cost functions for the main components are detailed in Section 4.3. For the diesel generator, the capital expenditure is evaluated according to Gnana [205], who provide a cost correlation based on the generator's rated electrical power. The reference cost corresponds to year 2017:

$$CAPEX_{DG}(\$) = 630 \cdot \text{Power(kW)} \quad (6.5)$$

$$LCOE = \frac{CAPEX + \sum_{i=1}^Y \frac{AC(i)}{(1+d)^i}}{\sum_{i=1}^Y \frac{E_L(i)}{(1+d)^i}} \quad (6.6)$$

The annual cost  $AC(i)$  included in the LCOE formulation accounts for maintenance expenses and, when applicable, fuel consumption:

$$AC(i) = O\&M + C_{\text{fuel}}(\$/\ell) \cdot \text{fuel}_{\text{consumption}}(\ell) \quad (6.7)$$

A system lifetime of 25 years is assumed for the LCOE calculation. Maintenance costs are set to 2% of the total investment, while the discount



rate is fixed at 5%, consistent with values commonly adopted in techno-economic assessments of comparable technologies [179–181].

Comparing the LCOE obtained across the three scenarios effectively determines, from a clean-slate perspective, which system configuration is the most economically favourable. This does not represent a repowering analysis in which the existing diesel generator is retained and additional components are integrated. Instead, it can be interpreted as evaluating alternative solutions for supplying electricity to a remote or underserved community, either one currently lacking access to electricity or one requiring the replacement of an ageing diesel generator. Accordingly, the analysis compares the cost of installing a new diesel generator with that of deploying various hybrid power-system configurations, without assuming that the existing diesel unit must be maintained.

To account for the uncertainty introduced by using cost correlations to estimate component capital costs, a Monte Carlo analysis was performed, following an approach that is commonly used in the literature [206, 207]. In this analysis, the investment cost of each component is randomly varied within a specific uncertainty range, for a total of 10,000 simulations. The uncertainty range is set in accordance with the AACE International Class 4 cost estimate classification [208, 209], which is appropriate for feasibility-stage studies. This classification prescribes an expected accuracy range of  $-25\% / +35\%$  around the deterministic estimate, interpreted as the P10–P90 confidence interval (i.e., an 80% confidence interval) per AACE convention. Specifically, a lognormal distribution is assumed for the capital cost of each component, the resulting right-skewed shape captures the tendency of capital costs to exhibit cost-overflow risk larger than downside potential. The uncertainty evaluation is carried out only for the configurations resulting from the optimisation process, during which the cost is directly computed through the proposed correlations. Moreover, the analysis accounts exclusively for the uncertainty associated with the CAPEX of the components included in the parametric inventory, and does not include other economic parameters, namely the diesel cost, the discount rate and the plant lifetime. The influence of these parameters is

instead investigated through a deterministic sensitivity analysis, in which each parameter is varied one at a time over a predefined set of values, applied to a subset of optimised configurations, as presented in detail in the Results section.

### 6.2.3 Life cycle Analysis

The environmental analysis is based on the Life Cycle Assessment (LCA) methodology, as standardized by ISO 14040 and ISO 14044 [210] [211]. The adopted approach is cradle-to-grave, so the system boundaries include the production and the decommissioning of the system. The functional unit for the analysis is the electrical kWh supplied to the end user. For the analysis, the Ecoinvent 3.10 database [212] and the OpenLCA software[213], were employed. The environmental impact, is computed through the Environmental Footprint 3.1 methodology. In particular, to facilitate the optimization process where the size of CAES equipments is changed, directly affecting the environmental impact, emphasis was put on generating a Parametric Life Cycle Inventory (pLCI). The creation of pLCI is composed by two main steps:

- Finding a reliable reference in order to construct an LCI of the considered component;
- Finding enough data to establish a relationship between the inventory and the size of the components.

This translates into relating the size of a component to a characteristic parameter (so-called life cycle parameter) and parameterizing the inventory in terms of it. In this work, the life cycle parameter is the mass for all the equipments, except for the PV installation; for the latter the installed peak power is considered. Firstly, an extensive market research has been conducted for each component; subsequently, one or more regressions were derived using a least squares methodology. The type of regression (i.e. linear, quadratic, power or logarithmic) was selected based on which model exhibited the highest coefficient of determination ( $R^2$ ). The obtained equations represent the relationship between mass and the scale of

the components. The reference LCI, collected for a specific size, is then scaled according to this correlations. All the reference LCI are available in Appendix-A. The scaled LCI describe materials and energy required and can be then converted into to environmental impact following the LCA procedure. Specifically, the parameter that describe the size of the components are: power for compressor, expander, electric motor and diesel generator; volume capacity for the thermal energy storage; nominal heat power for heat exchangers; kW of peak power for PV plant and both volume capacity and allowable pressure for air reservoir, for the last two components a slightly different approach is followed as described later on.

A Monte Carlo analysis analogous in framework to the one introduced for the economic assessment is also applied to the environmental evaluation, in order to quantify the uncertainty arising from the use of the mass correlations. As in the economic case, 10,000 simulations are performed. Whereas in the economic analysis the uncertainty range was set in accordance with the AACE Class 4 classification and modelled with a lognormal distribution, here the dispersion is instead derived directly from the underlying market data on which the mass correlations are built. Accordingly, a Gaussian distribution is assumed around the value provided by each correlation, with the standard deviation taken equal to the root mean square error (RMSE) between the correlation and the underlying market data, as defined in Equation (6.8). In this formulation, given a pair size-mass  $(x_i, y_i)$  derived from the market,  $f(x_i)$  denotes the corresponding mass approximation provided by the regression. As in the economic case, the analysis is restricted to the configurations resulting from the optimisation process, where the mass is computed directly through the proposed correlations, and produces both the mean value of each impact indicator and the corresponding 95% confidence interval. The uncertainty quantification accounts exclusively for the dispersion associated with the parametric mass inventory, and does not include other potential sources of uncertainty in the LCA, such as the allocation of mass to different materials or the characterisation factors adopted. This approach is commonly adopted for uncertainty quantification in life cycle

assessment studies [214, 215].

$$\sigma = RMSE = \sqrt{\frac{1}{N} \sum_{i=1}^n (f(x_i) - y_i)^2} \quad (6.8)$$

### 6.2.3.1 Compressor

The market analysis focused on reciprocating air compressors with multiple compression stages (typically 3–4) and operating pressures ranging from 50 to 200 bar. A broad power range was considered, extending to the upper applicability limits of this technology ( $\sim 1\text{MW}$ ). Figure 6.2 reports the market data together with the correlation derived from them and 95% confidence interval  $[-1.96\sigma, 1.96\sigma]$  of the associated gaussian distribution. A total of 238 products are considered derived from 10 of the most important compressor manufacturer (Boge, Atlas Copco, Dream Compressor, Denair, Burckardt, Sauer, Blaskmer, Sunceter, Bauer) [216–224]. Due to the scarcity of reliable data in the open literature, the LCI of an internal combustion engine was adopted as a reference, specifically the Ecoinvent dataset “*Internal combustion engine production, passenger car*”. This choice is motivated by the substantial equivalence of the two

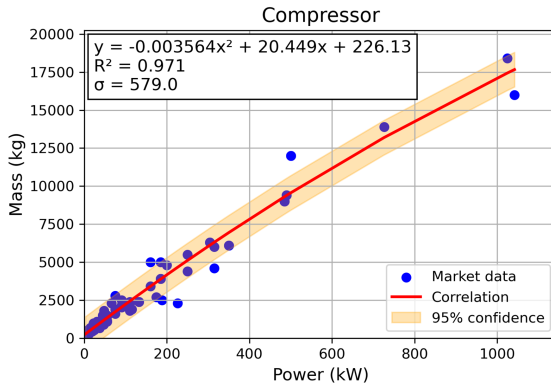


Figure 6.2: Compressor power - mass correlation

devices in terms of material composition and operating principle. Both systems share the same modular structure, consisting of pistons, cylinders, valves, a crankshaft, and a housing. Consequently, their manufacturing processes can be considered analogous.

### 6.2.3.2 Expander

As described in Section 6.2.1, the expander model can represent both scroll and screw expanders, with the transition from one technology to the other depending on the machine size. From an environmental assessment perspective, the market analysis was carried out for both categories. However, due to the lack of available data concerning both inventory parameters and mass-to-power ratios, scroll and screw expanders have been treated as compressors. Mass data are retrieved from catalogues of Atlas Copco and Fini Compressor for scroll compressor [217, 225] and Denair, Elang and Kaishan Compressor [219, 226, 227] for screw compressor. As shown in Figure 6.3, a single correlation can be defined that effectively approximates the mass of both machine types. Furthermore, manufacturer datasheet include not only the compressor itself but also the skid, the electric motor, and junction elements, further reducing

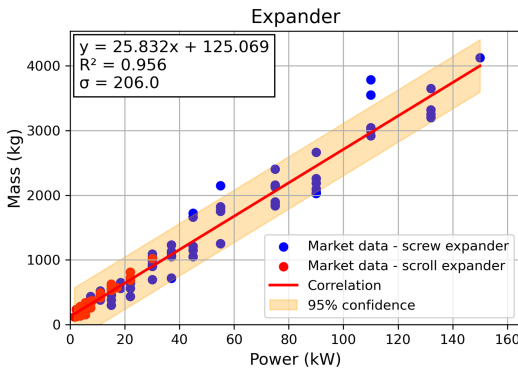


Figure 6.3: Expander power - mass correlation

the differences between the two technologies. For this reason, it was considered reasonable to assume their equivalence also in the impact assessment phase. The Ecoinvent process “4 kW screw compressor” has been chosen as the reference.

### 6.2.3.3 Heat Exchanger

Regarding the heat exchanger, since the system operates at high pressures, only data referring to steel components have been considered, data are derived from Pilan and Bowman catalogues [228, 229]. The nominal heat transfer rate was selected as the characteristic sizing parameter, as it provides a more robust correlation compared to a correlation based on heat transfer surface. Heat exchangers with the same surface but different mass flow rates, and consequently different tube diameters, exhibit significantly different masses. Figure 6.4 shows the obtained correlation. The reference LCI was taken from the work of [230], which describes a shell-and-tube heat exchanger employed in a micro-ORC system.

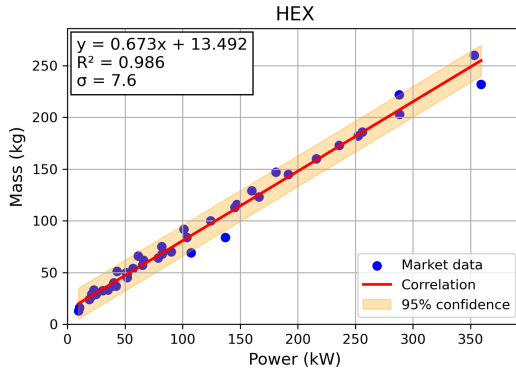


Figure 6.4: Heat exchanger power - mass correlation

#### 6.2.3.4 Thermal Energy Storage

To establish a volume–mass correlation for thermal energy storage (TES), the market analysis focused on hot water storage tanks, as substantially more data are available for this product and the storage medium does not affect the results. Market data were collected from the catalogues of ADCA Therm, IES, McKenna, Unical, Ekin, Tanpera, Kodsan, and AOSmith [231–237]. The resulting correlation is reported in Figure 6.5. For TES production modelling, a 270 L hot water tank, representative of small-scale solar thermal plants, was selected as reference. The LCI was derived from the work of Greening and Azapagic [238]. The storage medium was modelled separately, using the Ecoinvent dataset “lubricating oil” and assessing its environmental impact directly based on its total mass.

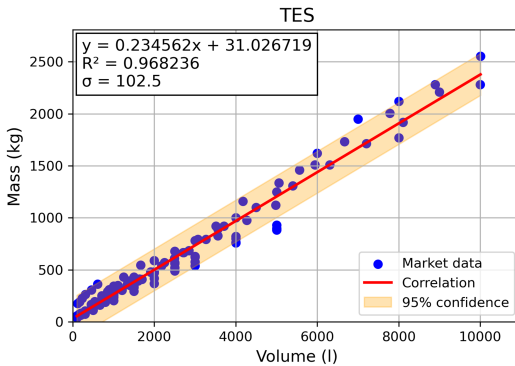


Figure 6.5: TES volume - mass correlation

#### 6.2.3.5 Electric Motor

While the reference inventory for expanders already includes the electric machine, in the case of compressors it was necessary to identify a suitable reference and develop a dedicated correlation, treating the motor as a separate component. Data were retrieved from Orange and OMEC

website [239, 240], considering only motors with efficiency class IE3 and a cast iron body, to ensure consistency with the selected reference case [241], Figure 6.6 display the market data and the derived correlation. The same power range explored for compressors was adopted for the motors. Although modelled independently, in the results section the impact for the electric motor are presented together with those of the compressor it drives, following the same approach applied to expanders.

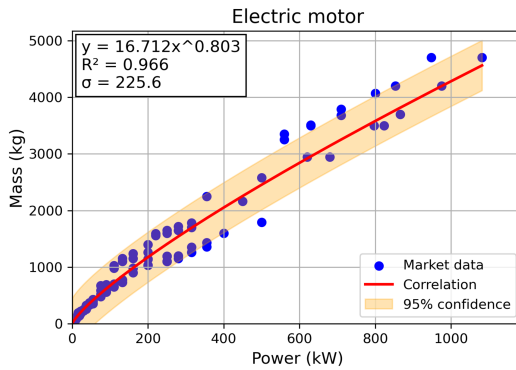


Figure 6.6: Electric motor power - mass correlation

#### 6.2.3.6 Diesel Generator

For diesel generators, a single regression is not sufficient to properly fit the market data. Therefore, the power range was divided into two domains, with a threshold at 50 kW. As shown in Figure 6.7, a logarithmic regression is applied for units below 50 kW, while a linear regression for higher powers. Although the two correlations are not continuous at the division point, the discrepancy between them is below 7%, which is acceptable for the purposes of this analysis. Market data are collected from the catalogues of Green Power Generators, Pramac, Yanmar, and CRI Fluid [242–245]. The reference LCI is based on a 5 kW diesel generator with a mass of 112 kg described in the study of Alsema [246]. The contributions from diesel



generator use phase are estimated using the relevant Ecoinvent processes: “Machine operation, diesel, < 18.64 kW, generators”, “Machine operation, diesel,  $\geq 18.64$  kW and  $\leq 74.57$  kW, generators”, and “Machine operation, diesel, > 74.57 kW, generators”, according to the generator’s nominal power. The inventories of these processes are scaled using the mass of diesel consumed as the scaling parameter. The diesel consumption (in kilograms) is calculated assuming a lower heating value (LHV) of 42,680 kJ/kg and an average generator efficiency of 0.35.

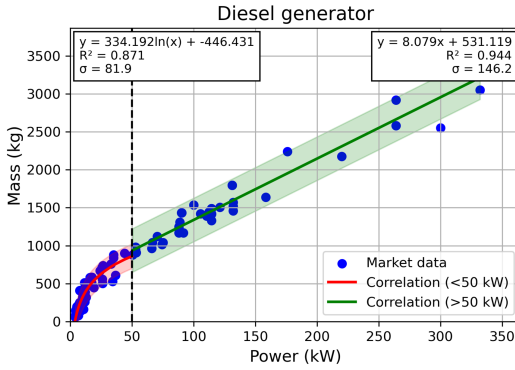


Figure 6.7: Diesel generator power - mass correlation

### 6.2.3.7 Air Vessels

For the air reservoir, no market-based correlation is introduced, but a different approach is adopted in order to explicitly capture the influence of both volume and maximum pressure on material demand. Vessel sizing is therefore carried out following the method proposed by Liu et al. [247] and summarized in Equations (6.9) to (6.11). In these equations,  $\delta$  denotes the vessel wall thickness, while  $\sigma_{\text{allow}}$  is the allowable tensile stress defined as  $\sigma_{\text{allow}} = \sigma_y / SF$ , with  $\sigma_y$  the yield strength. The considered reservoirs are standard 50 L steel bottles with an external diameter of 29.2cm. The selected material is 30CrMoE steel ( $\rho = 7850 \text{ kg/m}^3$ ,  $\sigma_y = 835 \text{ MPa}$ ),

and a safety factor of 2.5 is applied. The resulting material demand, and thus the environmental impact, depends on both storage volume and maximum pressure, both key parameters in CAES performance and optimization. The reference inventory for compressed air reservoir is derived from the study of Joshi [199].

$$\delta = \frac{D}{2} \cdot \left( 1 - \sqrt{\frac{\sigma_{all} - 1.3p_{max}}{\sigma_{all} + 0.4p_{max}}} \right) \quad (6.9)$$

$$V = \frac{\pi}{4} \cdot (D - 2\delta)^2 \cdot Ar \cdot D + \frac{\pi}{6} \cdot (D - 2\delta)^3 \quad (6.10)$$

$$m = \rho \cdot \left( \pi \cdot \delta \cdot (D - \delta) \cdot Ar \cdot D + \frac{\pi}{6} \cdot (D^3 - (D - 2\delta)^3) \right) \quad (6.11)$$

#### 6.2.3.8 PV system

To evaluate the environmental impact of the photovoltaic (PV) system, the Ecoinvent process “*Photovoltaic plant construction, 570 kWp, multi-Si, on open ground – GLO*” is selected as the reference. In line with the other components, the PV inventory was scaled during the optimization process, as its size is a key parameter in the analysis. However, unlike for the other components, the design and the life cycle parameters correspond for the PV, thus making the development of a specific correlation not necessary.

#### 6.2.4 Optimization method

In analogy with the approach adopted in the previous chapters, a component sizing optimization procedure is carried out relying on the EGO algorithm. The set of decision variables considered in the optimization includes those listed in Table 6.1. The bounds of the design space reported in the table refer to the base-case configuration. As discussed in more detail in the following subsection, in some scenarios the system is scaled to account for increased user demand; in such cases, the corresponding design-space limits are scaled accordingly.

Table 6.1: Decision variables and bounds for the sizing optimization of the PV-CAES-diesel system (base case)

| Parameter                       | Symbol     | Range                        |
|---------------------------------|------------|------------------------------|
| Compressor swept volume         | $V_{s,cc}$ | $500 \div 4000 \text{ cm}^3$ |
| Expander swept volume           | $V_{s,ex}$ | $10 \div 60 \text{ cm}^3$    |
| Pressure vessels volume         | $V$        | $10 \div 50 \text{ m}^3$     |
| Max air pressure inside vessels | $p_{max}$  | $40 \div 200 \text{ bar}$    |
| PV installed power              | $P_{PV,p}$ | $30 \div 200 \text{ kW}_p$   |

Concerning the objective function, the optimization was carried out from both economic and environmental perspectives. Three parameters were considered in the study:

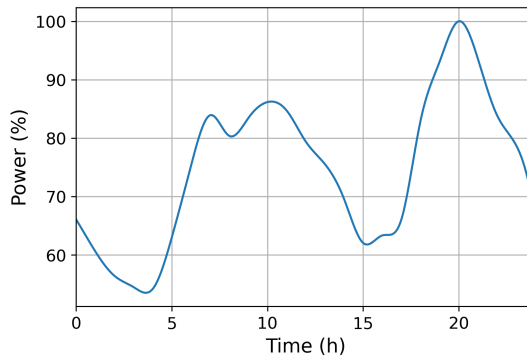
- the GWP ( $kgCO_{2,eq}/kWh$ ),
- the Single Score indicator ( $Pt/kWh$ ) (calculated according to the EF methodology),
- and the Levelized Cost of Electricity ( $\$/kWh$ )

The optimization procedure was applied to both Scenario 2 and Scenario 3. While Scenario 1 offers no room for improvement, as the user demand is fully supplied by the diesel generator, Scenarios 2 and 3 allow for an optimization process aimed at minimizing either environmental impact or cost. For a given user energy demand, the power supply can be provided through a combination of photovoltaic generation, CAES, and the diesel generator. Adjusting the system configuration through the sizing of individual components, thereby influencing system efficiency, environmental impact, cost, and the relative contribution of each energy source during operation.

### 6.2.5 User load demand and system location

The end-user is assumed to be a small residential energy community (e.g., a village). Therefore, the load profile considered corresponds to a statistical average of domestic electricity consumption [199]. As shown in Figure 6.8, the load exhibits typical household demand patterns, with

two daily peaks occurring in the morning and evening hours. The load is expressed in terms of maximum power demand. In the base case, the peak load is assumed to be 11 kW, corresponding to a total daily consumption of 200 kWh. As detailed in the following sections, in specific scenarios the maximum load is increased in order to assess the impact of system scaling, with analyses carried out up to 3 MWh of daily energy demand.



*Figure 6.8: End user power request*

To assess CAES performance and the potential economic and environmental benefits, seven different locations have been identified, all situated in countries where the adoption of ESS can support the electrification of rural areas and renewable energy integration. The criteria for selecting the case-study locations are based on three factors: the presence of off-grid areas that could benefit from such a system, the photovoltaic (PV) potential, and the diesel cost. In particular, both PV potential and diesel cost are classified into three ranges (low, medium, and high), and one representative nation is selected for each possible combination. The only exception is the case of low PV potential: here, only a nation with high diesel cost is considered, since it is unlikely that the hybrid system would be advantageous under conditions of both low PV potential and low diesel prices. Based on these considerations, the chosen locations are: Jordan, Cambodia, India, Botswana, Haiti, Romania and Zimbabwe. To consider

the impact of different climate zone (e.g. temperature) on the efficiency of photovoltaics panel, different value of photovoltaic performance ratio (PR) are considered based on the location, both photovoltaic potential and PR value are derived from Kawajiri et al. [248], diesel prices are obtained from the “Global Petrol prices” website [249]. Table 6.2 resume the characteristic of each installation. Hereafter, to more clearly describe the characteristics of each location, sites will be designated by an abbreviation reflecting both their photovoltaic potential and associated cost. For instance, Jordan will be denoted as HPV–LC (High Photovoltaic potential – Low Cost).

Table 6.2: Characteristics of the analyzed sites

| Location | PV potential<br>[kWh/kW] | Solar PR<br>[-] | Diesel cost<br>[\$/l] | Acronym |
|----------|--------------------------|-----------------|-----------------------|---------|
| Romania  | 1000-1200                | 0.75            | 1.609                 | LPV-HC  |
| Cambogia | 1200-1400                | 0.70            | 0.942                 | MPV-LC  |
| India    | 1200-1400                | 0.75            | 1.033                 | MPV-MC  |
| Belize   | 1200-1400                | 0.70            | 1.486                 | MPV-HC  |
| Jordan   | 1400-1600                | 0.75            | 0.905                 | HPV-LC  |
| Botswana | 1400-1600                | 0.75            | 1.167                 | HPV-MC  |
| Zimbabwe | 1400-1600                | 0.75            | 1.55                  | HPV-HC  |

6.3 Results and discussion

In this section, the results of the study are presented. First, the three scenarios are compared for each location, in term of environmental aspect. For the sake of brevity, only the results of one of the six studied installations are reported in detail, specifically the one with the lowest greenhouse gas emissions. The outcomes of the other case studies are summarized concisely. For the lower-GWP scenario a comparison with the outcomes obtained when the objective function is expressed in terms of single-score assessment is also presented. Finally, the economic results are described.

### 6.3.1 Comparison of Different Scenarios under GWP Minimization

For each case location, the configuration with the lowest greenhouse gas emissions is obtained, for both scenario 2 and 3, using the algorithm described in Section 6.2.4. Figure 6.9 shows the results obtained in terms of equivalent CO<sub>2</sub> emission per kWh, with a focus on the third scenario. Scenario 1, identical across all locations, results in 0.996 kg CO<sub>2</sub>eq/kWh. The addition of a PV system reduces emissions to approximately 0.598 kg CO<sub>2</sub>eq/kWh, while implementing CAES in Scenario 3 further lowers emissions by roughly 30%, ranging from 0.144 to 0.247 kg CO<sub>2</sub>eq/kWh depending on location. Figure 6.10 illustrates the share of energy supplied by diesel, PV, and CAES, along with CO<sub>2</sub>eq emissions separated into use

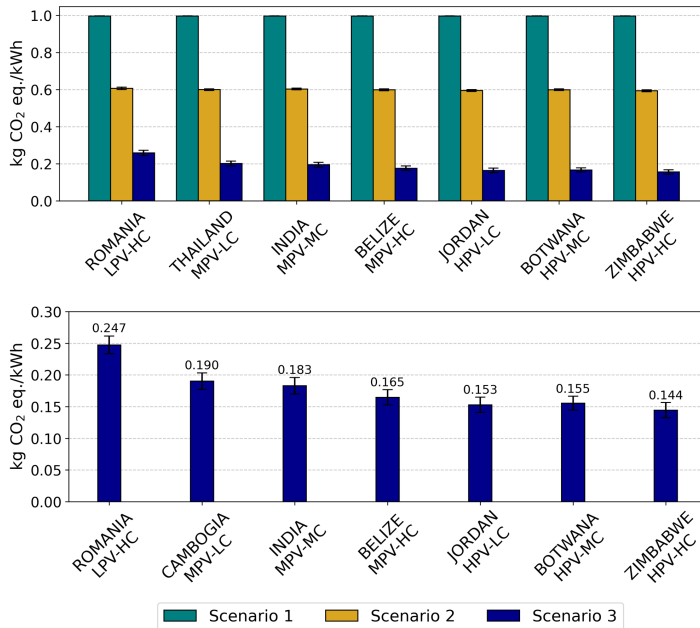


Figure 6.9: Global warming potential for the three scenarios

and end-of-life (EoL) and construction phases, for the sites with the highest (Romania) and lowest (Zimbabwe) emissions. These results indicate that the reduction in emissions from Scenario 1 to Scenario 3 is primarily due to the decrease in direct diesel generator operation. In Zimbabwe, for instance, the diesel contribution to total energy supply decreases from 100% in Scenario 1, to 54.5% in Scenario 2, and further down to only 1.7% in Scenario 3. Although higher system complexity increases construction-related emissions, as illustrated in Figure 6.10(b), the sharp decline in operational emissions ultimately results in lower overall CO<sub>2</sub>eq emissions. By comparing the results reported in the graphs for the two locations and the optimal configuration sizing in Table 6.3, it becomes clear why Scenario 2 exhibits only minor geographical variation, whereas the configuration

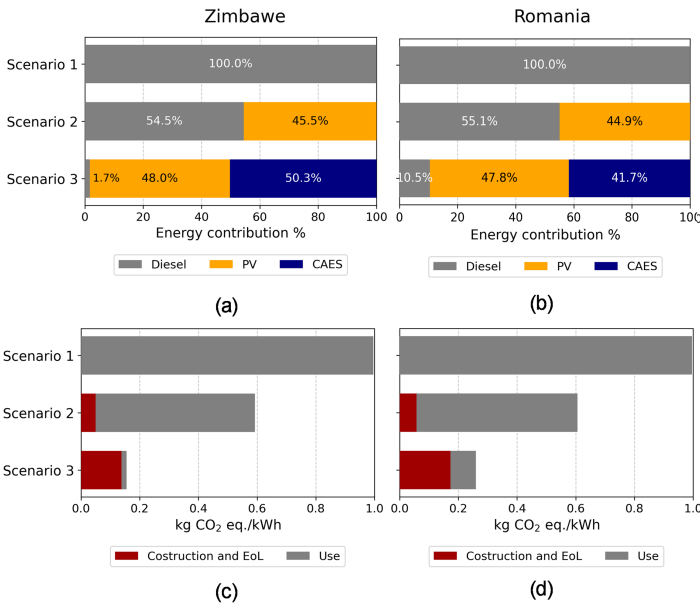


Figure 6.10: Energy contribution and GWP breakdown for the three scenarios at the HPV-HC and LPV-HC site

including CAES shows more pronounced differences. The two sites have markedly different photovoltaic potentials: the LPV site requires a larger PV capacity, and while this difference remains moderate in Scenario 2, it becomes substantial in Scenario 3, where surplus solar energy is stored in the CAES. Under low-PV conditions, renewable coverage remains lower despite a larger installed PV capacity, increasing both construction and operation related impacts. This underscores the central role of the PV system in shaping overall environmental performance, as further discussed in the following sections. The bar charts in Figure 6.9 also report the uncertainty associated with the adopted pLCI method, quantified as described in Section 6.2.3. For the first two scenarios, uncertainty is negligible because impacts are dominated by fuel combustion. In Scenario 3, where construction and end-of-life processes account for most GWP emissions, uncertainty is higher but remains within an acceptable range ( $\pm 3.5\%$  of the total). Furthermore, Table 6.3 reports a round-trip efficiency of approximately 35%. Although relatively low, this value is consistent with documented performance of micro-CAES systems based on conventional technologies, which do not employ advanced concepts such as isothermal machines still under development [26]. The use of reciprocating compressors and volumetric expanders, characterized by modest efficiencies and significant thermal losses, largely explains this outcome. As a consequence, the low RTE necessitates a substantial oversizing of the PV array, which increases markedly when transitioning from the PV–Diesel to the PV–Diesel–CAES configuration and, as will be shown, constitutes a major contributor to the overall environmental burden of the system.

To further compare the environmental impact of the three different scenarios, Figure 6.11 reports the absolute value of four relevant midpoint-level impact in the three scenarios, while Figure 6.12 shows the value of all the midpoint indicator normalized over the maximum. Figure 6.11a–b highlight the categories energy resource depletion (ER) and particulate matter formation (PM), which decrease by about 40% from Scenario 1 to Scenario 2. In Scenario 3, reductions are even more pronounced,



Table 6.3: Minimum GWP configuration for Zimbabwe and Romania

|          |            | $V_{s,c}$<br>[cm <sup>3</sup> ] | $V_{s,ex}$<br>[cm <sup>3</sup> ] | $p_{\max}$<br>[bar] | $V$<br>[m <sup>3</sup> ] | $P_{PV,p}$<br>[kW <sub>p</sub> ] | RSS  | RTE  |
|----------|------------|---------------------------------|----------------------------------|---------------------|--------------------------|----------------------------------|------|------|
| Zimbabwe | Scenario 2 | -                               | -                                | -                   | -                        | 38                               | 45.5 | -    |
|          | Scenario 3 | 2150                            | 17                               | 91                  | 23.1                     | 74                               | 98.8 | 35.7 |
| Romania  | Scenario 2 | -                               | -                                | -                   | -                        | 43                               | 44.9 | -    |
|          | Scenario 3 | 2365                            | 19                               | 103                 | 18.4                     | 89                               | 89.5 | 34.5 |

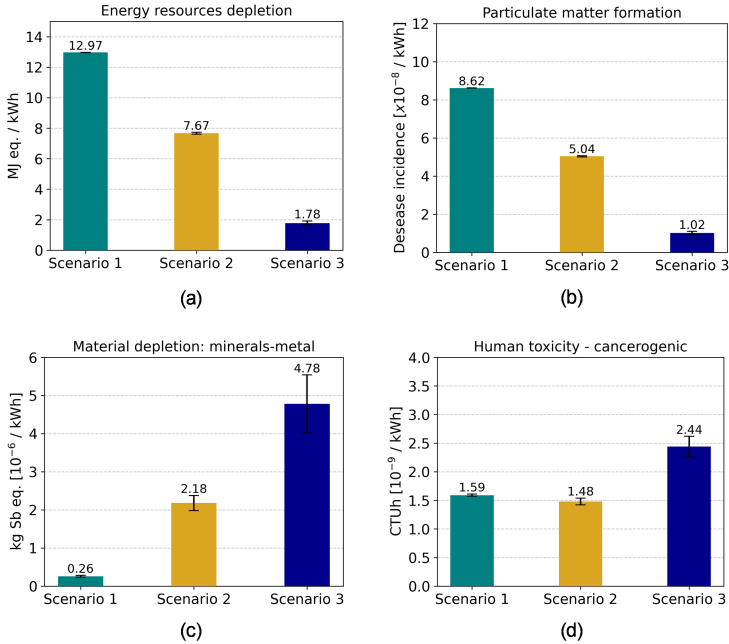


Figure 6.11: Comparison of relevant mid point for the three scenarios at the HPV-HC site: (a) Energy resource depletion - (b) Particulate matter formation - (c) Material depletion - (d) Human toxicity carcinogenic

reaching 86% and 88%, respectively. These improvements are mainly due to the sharp decrease in diesel use; although non-renewable energy demand increases during construction, diesel consumption during operation

remains the dominant factor. Conversely, Figure 6.11c–d show opposite trends. Mineral resource depletion (MR) rises steadily from Scenario 1 to Scenario 3 as system components increase, while human toxicity - carcinogenic (HTC) remains nearly unchanged between Scenarios 1 and 2 but almost doubles in Scenario 3. This effect is linked to metal processing, as steelmaking in electric arc or basic oxygen furnaces generates large amounts of toxic slag requiring treatment and disposal. Once again, the uncertainty introduced by the proposed correlations leads to negligible values in the first two scenarios. In the third scenario, it remains negligible for the categories where the impact is associated with combustion (ER and PM), while it becomes more relevant for HTC and MR. In particular, the uncertainty increases significantly for the MR category, which is more closely related to the mass of individual components. In this case, for Scenario 3, the uncertainty reaches approximately  $\pm 15\%$ , indicating that the estimation cannot be considered fully reliable. Nonetheless, such elevated uncertainty values are observed exclusively for this category and in this specific scenario. A more precise estimation would be required for a detailed assessment of this category; however, overall, the estimated impacts are not strongly influenced by correlation uncertainty.

From the radar chart reported in Figure 6.12, it can be observed that Scenario 2 shows an intermediate behavior, whereas Scenarios 1 and 3 exhibit opposite trends for most impact categories. In Scenario 3, emissions increase in all categories where material processing plays a major role. These include freshwater eco-toxicity (FET), freshwater eutrophication (FE), human toxicity carcinogenic and non-carcinogenic (HTC, HTNC), ionizing radiation (IR), and mineral resource depletion (MR). Additional increases are also observed in categories directly linked to the larger photovoltaic installation, such as land use (LU) and water use (WU). On the other hand, Scenario 1 shows higher values in all categories strongly affected by fossil fuel combustion, such as acidification (AT), climate change (GWP), energy resource depletion (ER), marine and terrestrial eutrophication (ME, TE), ionizing radiation (IR), ozone depletion (OD), particulate matter formation (PM), and photochemical

oxidant formation (POF).

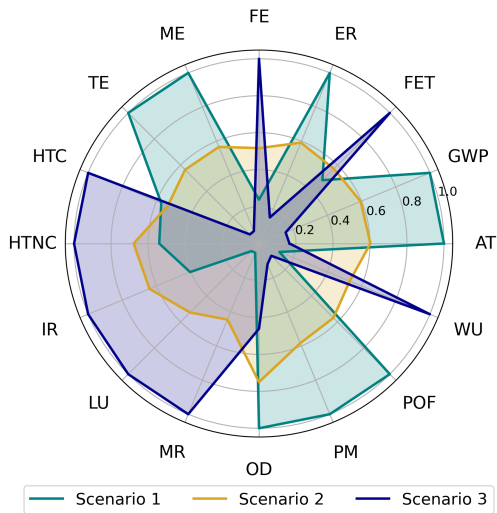


Figure 6.12: Mid point comparison for the three scenarios at the HPV-HC site

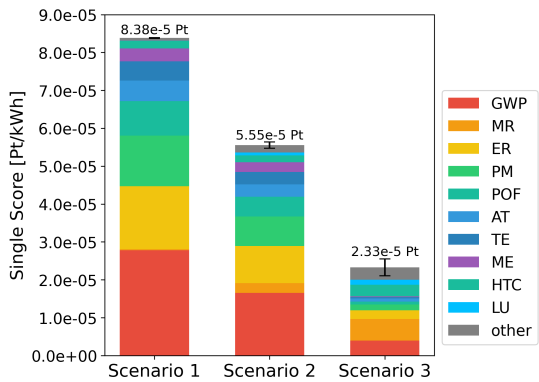


Figure 6.13: Comparison of single score for the three scenarios at the HPV-HC Site

To summarize the overall environmental performance across all impact categories, the results of the single score analysis are presented in Figure 6.13, highlighting the contribution of the main categories. The single score confirms the benefits of storing solar energy in the form of compressed air, even though not all categories are improved by the installation of the storage system. The overall impact decreases from  $8.38 \cdot 10^{-5}$  Pt/kWh in Scenario 1 to  $5.55 \cdot 10^{-5}$  Pt/kWh in Scenario 2, and further down to  $2.33 \cdot 10^{-5}$  Pt/kWh in Scenario 3. Figure 6.13 also shows that in Scenarios 1 and 2 the dominant contributions come from GWP and resource depletion, both strongly linked to diesel consumption. These categories experience the most pronounced reductions when moving from Scenario 1 to Scenario 3 via Scenario 2. In the final configuration (Scenario 3), the contribution of the different impact categories appears more balanced, with a relative increase in the role of categories associated with material-intensive components, such as mineral resource depletion.

Focusing on Scenario 3, Figure 6.14 reports the percentage breakdown of impacts across the midpoint categories. The graph clearly shows that a substantial portion of life cycle emissions is attributable to the photovoltaic installation. For climate change, approximately 70% of the total

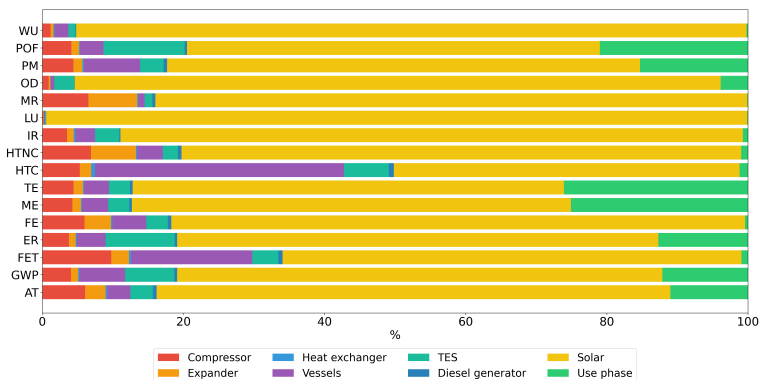


Figure 6.14: Percentage breakdown of impacts for midpoint categories

impact arises from the production and disposal of silicon panels. Climate altering emissions during the use phase depend on the amount of diesel consumed, with almost all burdens originating from direct combustion. A smaller contribution is linked to diesel production and transport, as well as lubricating oil. The impact of the thermal storage unit derives mainly from mineral oil, about half of its greenhouse gas emissions are associated with the end-of-life stage, as the oil is typically incinerated as hazardous waste, directly releasing greenhouse gases into the atmosphere. The relative contribution of components varies considerably across other impact categories. For terrestrial acidification, the dominant contribution is from the PV installation, particularly from multi-Si cell production. Human toxicity and mineral resource depletion are instead driven by the construction of the CAES system, due to the substantial material requirements. Within human toxicity, the carcinogenic impact is largely attributable to steel production, whereas in the non-carcinogenic case, steel-related emissions are less relevant, and the main contribution shifts to PV cell manufacturing. The fossil energy use category shows the highest burden in the use phase, reflecting diesel consumption, although metal-working remains non-negligible. Particulate matter formation is again dominated by the PV system, with silicon processing as the main hotspot. Finally, land use and water use indicators are almost entirely controlled by PV installation, with land occupation from solar panels representing nearly 100% of the total. In general terms, Figure 6.14 shows that, in Scenario 3, the majority of the environmental impact is attributable to the construction phase of the photovoltaic panels. This result might appear to contradict what is commonly reported in the literature, where LCA studies of CAES systems indicate that the main contribution to the environmental impact arises from the operational phase. However, this apparent discrepancy stems from the definition of the system boundaries adopted in the present LCA study, which include the storage unit, the photovoltaic system, and the diesel generator within the system boundary. Consequently, the results characterize the entire plant responsible for supplying electricity to the end user. If the environmental effects of the

CAES unit alone were to be assessed, all emissions associated with the photovoltaic system would be attributed to the operational phase. Under this assumption, the greenhouse gas emissions per kWh of the CAES would amount to 0.212 kg CO<sub>2</sub> eq., a higher value than that obtained for Scenario 3, which represents a combined impact associated with the combined operation of the compressed air storage, diesel generator, and photovoltaic generation systems. This value (i.e., 0.212 kgCO<sub>2</sub>eq/kWh) is consistent with values reported in the literature for CAES systems operating under electricity mixes with moderate carbon intensity. For electricity mixes with comparable carbon intensity, similar greenhouse gas emission intensities are also reported in the literature for battery-based energy storage systems [51, 52]. However, a detailed and systematic comparison between CAES and battery technologies is beyond the scope of this work and would require a harmonized life cycle assessment framework, representing a clear direction for future research.

### 6.3.2 Comparison between GWP and Single Score Minimization Results

In this section, a comparison is presented between the results obtained when the optimization process is performed using the Single Score (SS) as the objective function instead of minimizing GWP. For brevity, results are reported only for the Zimbabwe case study, already analyzed in more detail in the previous section. This approach provides insights into how the optimal configuration evolves when all midpoint indicators are considered simultaneously. Although GWP remains one of the most widely used indicators in environmental assessments, this comparison helps clarify how system design may change when broader environmental aspects are accounted for. The results, summarized in Table 6.4, show that the two configurations differ only marginally in terms of both system sizing and environmental impact. Minor deviations are nevertheless observed: in the SS-based solution, the air storage capacity and the PV subsystem are slightly reduced, while machine displacement remains essentially unchanged. This adjustment results in a lower energy contribution from

Table 6.4: Minimum GWP and minimum SS configuration

|         | $V_{s,c}$<br>[cm <sup>3</sup> ] | $V_{s,ex}$<br>[cm <sup>3</sup> ] | $p_{\max}$<br>[bar] | $V$<br>[m <sup>3</sup> ] | $P_{PV,p}$<br>[kW <sub>p</sub> ] | GWP<br>[ $\frac{\text{kg CO}_2 \text{ eq.}}{\text{kWh}}$ ] | SS<br>[Pt/kWh]        |
|---------|---------------------------------|----------------------------------|---------------------|--------------------------|----------------------------------|------------------------------------------------------------|-----------------------|
| Min GWP | 2150                            | 17.0                             | 91                  | 23.1                     | 74                               | 0.144                                                      | $2.33 \times 10^{-5}$ |
| Min SS  | 2045                            | 16.3                             | 100                 | 18.2                     | 67                               | 0.152                                                      | $2.26 \times 10^{-5}$ |

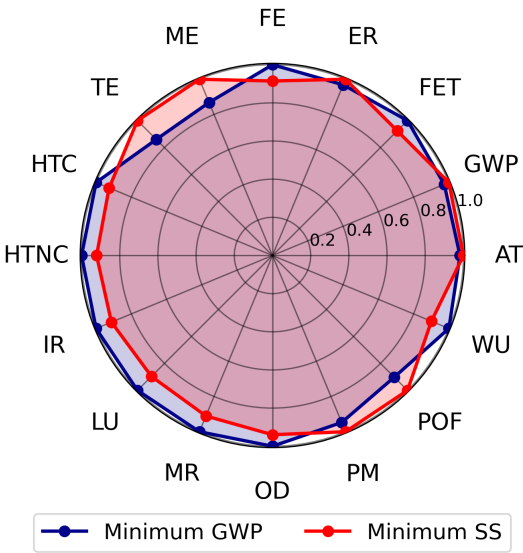


Figure 6.15: Mid point comparison for minimum GWP and minimum SS configuration

CAES and a consequent increase in CO<sub>2</sub> emissions from diesel generation. Specifically, GWP increases from 144 to 152 gCO<sub>2</sub>eq/kWh, while the Single Score decreases from  $2.33 \times 10^{-5}$  to  $2.26 \times 10^{-5}$ .

The radar chart in Figure 6.15 compares the normalized values of all midpoint indicators with respect to their maximum. It clearly shows that differences between the two configurations remain limited, with variations generally within 10% of the maximum value. When the SS is minimized,

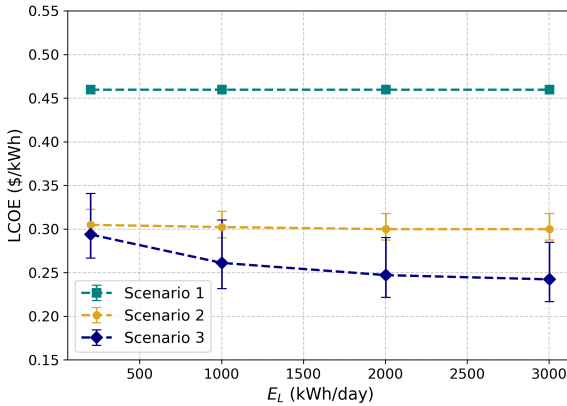
the reduction in installed PV power and storage volume slightly decreases the impacts of categories mainly associated with metal processing, such as ecotoxicity, eutrophication, and human toxicity; conversely, it leads to an increase in categories that are strongly influenced by direct fuel combustion emissions. Overall, the two optimization approaches, converge towards very similar system configurations. This behavior is largely explained by the relatively high weight attributed to climate change (approximately 20%) in the EF normalization and weighting set. The configuration that minimizes climate change emissions therefore closely corresponds to the one that minimizes the aggregated indicator, reinforcing the robustness of the GWP-based optimization despite the uncertainty of the SS. Importantly, the conclusions drawn in the previous section regarding the comparison of the three scenarios and the contribution of the different CAES components remain valid even if the optimization is performed with the Single Score as the objective function.

### 6.3.3 Economic assessment results

Figure 6.16 reports the LCOE for the three scenarios in the most promising location (Zimbabwe, characterised by high photovoltaic potential and elevated diesel costs) at varying system scales. The different scales, expressed in terms of total daily user energy demand, were derived by multiplying the base load profile by a specific coefficient: from left to right, the plotted data correspond to peak loads of 11, 55, 110 and 160 kW, respectively. The markers indicate the LCOE obtained from the optimisation process using the deterministic estimates provided by the cost correlations, while the error bars represent the 80% confidence interval (P10–P90) derived from the Monte Carlo analysis described in Section 6.2.2, in accordance with the AACE Class 4 accuracy range. While the LCOE for Scenarios 1 and 2 remains nearly constant with increasing system size, Scenario 3 exhibits a decreasing trend. This behaviour is explained by the fact that, whereas the costs of diesel generators and PV modules scale linearly, the cost functions describing the CAES equipment are non-linear, thereby benefiting from economies of scale. Scenario 1



results in the highest LCOE (0.459 \$/kWh). At the base system size the configuration including CAES storage shows a slightly lower LCOE than Scenario 2 (0.294 \$/kWh for the former and 0.299 \$/kWh for the latter); however, due to the decreasing trend of Scenario 3, the integration of CAES becomes clearly more cost-effective at larger sizes, with the LCOE of Scenario 3 declining to 0.244 \$/kWh at 2 MWh/day and gradually approaching an asymptotic value at larger scales. The uncertainty band widens progressively from Scenario 1 (practically negligible) to Scenario 2 ( $-4\%$  /  $+6\%$ ) and Scenario 3 ( $-10\%$  /  $+17\%$ ), reflecting the growing weight of CAPEX as the share of fuel-related OPEX decreases. The economic advantage of the storage-based configurations over Scenario 1 is unequivocal across all load sizes, with no overlap between the corresponding uncertainty bands. The comparison between Scenarios 2 and 3 is more nuanced: at the smallest size ( $E_L = 200$  kWh/day) the two intervals overlap substantially and the corresponding LCOE values are close, so that no firm economic preference between the two solutions can be established at this scale. As system size increases, however, the LCOE of Scenario 3 moves progressively below that of Scenario 2 and the overlap



*Figure 6.16: LCOE for the three scenarios in HPV-HC site at different system scale*

reduces, with the P90 of Scenario 3 falling below the P10 of Scenario 2 at the largest size considered. The cost advantage of the storage-based solution thus becomes increasingly robust as the system size grows, while remaining sensitive to CAPEX uncertainty.

These results indicate that micro-scale compressed air systems have limited cost-effectiveness for very small off-grid installations, but become increasingly competitive as the system scale grows. In the largest configuration analysed, the reciprocating compressor included in the system has a rated capacity of approximately 550 kW, which corresponds to the upper limit at which this technology is typically employed; for larger system sizes, the adoption of centrifugal compressors could represent a more suitable solution.

**Sensitivity to economic parameters.** The results of the sensitivity analysis to diesel price, discount rate, system lifetime and maintenance cost are reported in Figures 6.17 and 6.18 for the optimised configurations of the minimum and maximum-size systems, respectively. The three scenarios respond differently to variations in these parameters, consistently with their distinct cost structures: Scenario 1 (diesel-based) combines low CAPEX with high OPEX, Scenario 2 (with PV integration) lies in an intermediate position, while Scenario 3 (with CAES storage) exhibits the opposite structure, with high CAPEX and limited OPEX. Being fuel-intensive, Scenarios 1 and 2 display steep slopes against diesel price variations, whereas Scenario 3 remains nearly flat across the considered range. The opposite behaviour is observed for the other three parameters: Scenarios 1 and 2 are largely insensitive to maintenance rate, system lifetime and discount rate, while Scenario 3 shows appreciable variations, especially with respect to the discount rate. This is consistent with the fact that a CAPEX-intensive configuration is intrinsically more sensitive to the discount rate and to the assumptions on useful life and annual operating costs associated with the initial investment.

Among the parameters considered, the diesel price and the discount rate exert the strongest influence on the overall economic competitiveness

of the alternatives. The storage-based configuration is particularly sensitive to variations in the discount rate, yet it proves remarkably resilient to fluctuations in fuel cost. This is a relevant feature in the current global context, in which diesel prices tend to follow a structurally increasing trend: under such conditions, the storage-based solution becomes progressively more competitive as fuel cost rises. Extending the system lifetime likewise benefits Scenario 3 significantly, as the amortisation of CAPEX over a longer period reduces the LCOE appreciably, while Scenarios 1

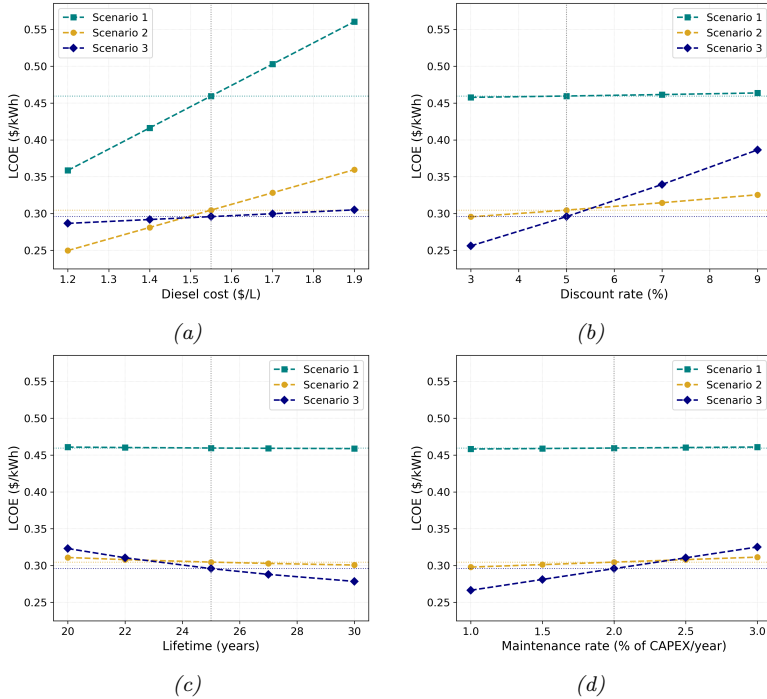


Figure 6.17: Sensitivity analysis of the LCOE for the minimum analyzed system size: influence of (a) diesel fuel cost, (b) discount rate, (c) system lifetime and (d) maintenance rate

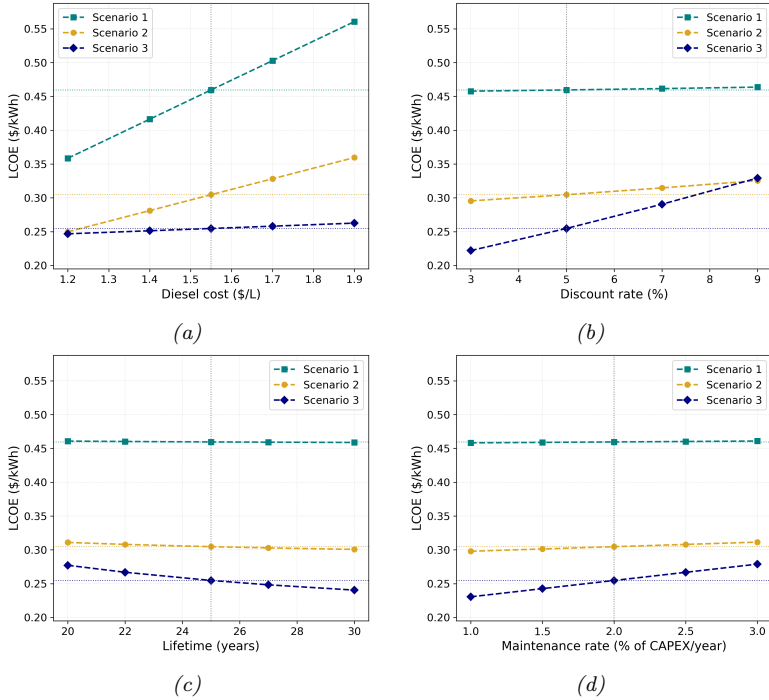


Figure 6.18: Sensitivity analysis of the LCOE for the maximum analyzed system size: influence of (a) diesel fuel cost, (b) discount rate, (c) system lifetime and (d) maintenance rate

and 2 remain essentially unaffected, their LCOE being dominated by fuel procurement. An analogous reasoning applies to the maintenance cost, assumed equal to 2% of CAPEX per year: its weight is appreciable only in capital-intensive configurations, and therefore mainly in Scenario 3.

It is also worth noting that, although a discount rate of 5% may be considered optimistic for off-grid systems deployed in remote or developing contexts (typically associated with higher country risk), in the maximum-size configuration an increase up to 9% does not undermine the

competitiveness of Scenario 3. Conversely, in the minimum-size configuration the LCOE of Scenario 3 becomes higher than that of Scenario 2 when some parameters are varied, particularly at low diesel prices or high discount rates. Overall, the uncertainties associated with the economic analysis are non-negligible; nevertheless, the present study provides meaningful indications regarding the potential economic feasibility of the storage-based system. The results also highlight that a more accurate estimation of the economic performance would require a refined characterisation of the investment costs of the individual components, together with a detailed assessment of the installation context, which influences both the diesel price and the discount rate.

**GWP vs LCOE-optimal configurations.** Table 6.5 summarises the design and performance parameters of the configurations that minimise GWP and LCOE for Zimbabwe, considering both the smallest and largest system sizes. A clear distinction emerges between the outcomes associated with the two objective functions, in terms of both design choices and system performance. When cost minimisation is targeted, the optimisation yields smaller photovoltaic and storage capacities compared with the GWP-minimising case, at the expense of lower renewable self-sufficiency. The optimisation also identifies different storage-pressure values that minimise environmental impact ( $\sim 90$  bar) and system cost ( $\sim 117$  bar).

Moreover, while the GWP-optimal configurations scale almost linearly with the user's energy demand, the economically optimal ones exhibit a more than linear increase in component size. This behaviour is driven by

*Table 6.5: Comparison of optimized configurations for Zimbabwe considering either GWP or LCOE as the objective function, for the minimum and maximum analyzed system sizes.*

| Objective function | E <sub>request</sub><br>[kWh/daily] | V<br>[m <sup>3</sup> ] | V <sub>s,c</sub><br>[cm <sup>3</sup> ] | V <sub>s,ex</sub><br>[cm <sup>3</sup> ] | P <sub>PV,p</sub><br>[kW <sub>p</sub> ] | p <sub>max</sub><br>[bar] | RTE<br>[%] | RSS<br>[%] | GWP<br>[ $\frac{\text{kg CO}_2 \text{ eq.}}{\text{kWh}}$ ] | LCOE<br>[ $\frac{\$}{\text{kWh}}$ ] |
|--------------------|-------------------------------------|------------------------|----------------------------------------|-----------------------------------------|-----------------------------------------|---------------------------|------------|------------|------------------------------------------------------------|-------------------------------------|
| GWP                | 200                                 | 23.1                   | 2150                                   | 17                                      | 74                                      | 91                        | 35.7%      | 98.8%      | 0.144                                                      | 0.322                               |
|                    | 3000                                | 353.2                  | 32496                                  | 250                                     | 1146                                    | 93                        | 35.4%      | 98.8%      | 0.140                                                      | 0.287                               |
| LCOE               | 200                                 | 13.3                   | 1640                                   | 15                                      | 69                                      | 117                       | 35.1%      | 93.9%      | 0.174                                                      | 0.294                               |
|                    | 3000                                | 203.7                  | 26106                                  | 226                                     | 1018                                    | 117                       | 35.4%      | 94.4%      | 0.167                                                      | 0.244                               |

the nonlinear cost scaling of several components, which show sublinear cost growth with capacity. As a result, larger systems achieve slightly higher renewable self-sufficiency (RSS increases from 93.9% to 94.4%) while simultaneously reducing the cost of electricity. The same nonlinear scaling explains why the sensitivity of GWP to system size is so modest, approximately 3% reduction in Scenario 3 from minimum to maximum size. Although some size–mass correlations are sublinear, the dominant contribution to the environmental impact comes from the photovoltaic system, whose mass, and hence impact, scales linearly with size, and the disparity among the three scenarios is large enough that size-related variations do not alter the overall ranking. This trend is further reflected in Figure 6.19, which presents the investment cost breakdown for the LCOE-minimising configurations. A reduction is observed in the relative contribution of components governed by nonlinear cost functions: particularly the heat exchangers, followed by the compressor and expander. Conversely, the largest cost contributions arise from the photovoltaic system and the pressure vessels, which represent the most expensive component of the CAES system. Within the range of system sizes analysed,

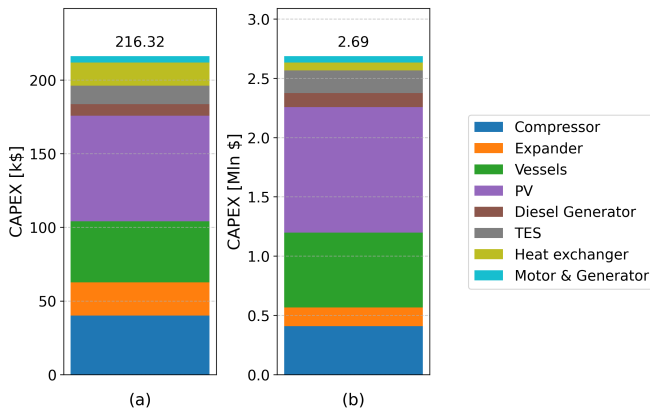


Figure 6.19: CAPEX breakdown for configurations that minimize the LCOE for the minimum (a) and maximum (b) analyzed system sizes

the significant increase in storage volume also suggests that alternative air storage solutions could become technically preferable: relying on a large number of small, standardised pressure vessels may no longer represent the most convenient option from a plant design perspective, and fewer vessels with larger individual volumes could reduce system complexity and balance-of-plant requirements. In the present work, however, no change in storage technology is assumed, and the same vessel design adopted at the minimum size is maintained across all configurations to ensure a coherent comparison, while acknowledging that storage technology selection may influence both cost and system layout as volumetric requirements increase.

**Geographical dependence.** Figure 6.20 shows the economic results obtained for the maximum system size across all seven locations, including the uncertainty bar corresponding to the CAPEX uncertainty according to AACE Class 4. An economic benefit from CAES integration is observed only for the MPV-HC and LPV-HC locations (Belize and Romania), which are also characterised by high fuel prices. These findings highlight the strong dependence of system performance on diesel cost: the implementation of storage provides a benefit only in regions with high fuel prices, where minimising diesel generator usage is advantageous, while in countries with lower fuel costs CAES implementation may not be economically advisable. PV installation, on the other hand, remains beneficial in all cases when compared to full diesel dependency. The limited competitiveness of compressed air storage systems is mainly attributable to their low round-trip efficiency (approximately 35–36%), which results in an excessively oversized PV system. Comparing these conclusions with the results of Section 6.3.1, the asymmetry between the two perspectives is clear: while solar energy storage offers environmental benefits irrespective of geographic location, the economic viability of CAES is not guaranteed by the present configuration and remains strongly dependent on local economic factors.

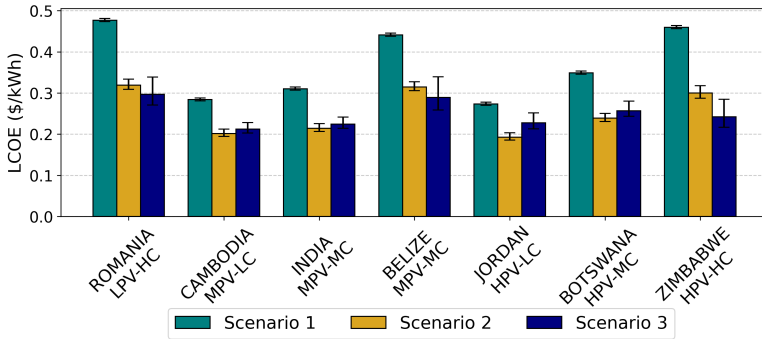


Figure 6.20: LCOE for the three scenarios in different geographic sites

## 6.4 Conclusion

This chapter represents a first point of closure of the thesis objectives, as it applies the modelling, optimization, and assessment frameworks developed and validated in the previous chapters to a comprehensive system-level evaluation. Specifically, a parametric Life Cycle Assessment and techno-economic analysis is performed for an hybrid PV–diesel–CAES systems for off-grid power generation across different geographic contexts. The results provide insights into the environmental and economic implications of implementing compressed air energy storage at small scales. Through the parametric approach, the work investigated how design parameters, local solar potential, and fuel costs influence both sustainability and cost-effectiveness.

The main findings can be summarized as follows:

- The study introduces size–power correlations enabling the scaling of reference inventories to better reflect the behavior of commercial components. The uncertainty associated with using these correlations to estimate actual component dimensions was explicitly accounted for and found to be acceptable for the scope of the present analysis. These correlations establish a parametric foundation for future LCA-based assessments.



- CAES integration consistently lowers CO<sub>2</sub> emission. Compared with the diesel-only system (0.996 kg CO<sub>2</sub>eq/kWh), the hybrid PV–diesel configuration reduces emissions by approximately one third, while the PV–diesel–CAES system achieves reductions to as low as 0.144 kg CO<sub>2</sub>eq/kWh.
- The sequential addition of photovoltaic generation and CAES storage results in an overall improvement in environmental performance, both in terms of Global Warming Potential and aggregated single-score indicators. Nonetheless, higher impacts are observed in categories linked to metal production and raw material extraction (e.g., mineral resource depletion, human toxicity).
- In the PV–diesel–CAES configuration, the photovoltaic subsystem emerges as the dominant contributor to total environmental impact, primarily due to the oversizing required to compensate for the limited round-trip efficiency of CAES.
- From an economic perspective, CAES integration is hardly cost-competitive at micro scales ( $\sim 10$  kW), with profitability mainly driven by increased system size and, especially, by installations in regions with high diesel prices, where the fuel substitution benefits outweigh the additional capital expenditure. In most other contexts, the study highlights a clear mismatch between environmental improvement and economic viability under current technological and market conditions. It is worth noting, however, that the current global economic context is characterized by a continuous increase in fossil fuel prices. This trend could make the use of PV-CAES based systems economically attractive in a growing number of installation sites.

Overall, CAES demonstrates strong potential for reducing greenhouse gas emissions in off-grid applications and represents a viable pathway for decarbonizing diesel-dependent communities. However, its economic viability remains constrained at small scales and highly sensitive to local

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conditions. The pronounced dependence on diesel price and system size emphasizes the need for context-specific techno-economic and environmental assessments prior to deployment.



## Chapter 7

# System Scaling and Integration of Radial Turbomachinery in Small-Scale Off-Grid Systems

Building on the results discussed in the previous chapter, this chapter extends the analysis to systems of slightly larger scale. In particular, the focus is on investigating the effects of transitioning from reciprocating volumetric machines to radial turbomachinery for both the compression and expansion processes. The objective is to assess whether the adoption of radial machines in small-scale and off-design operating conditions can provide advantages in terms of efficiency and economic feasibility.

To this end, the modeling approach adopted for radial turbomachinery is introduced, with particular emphasis on the regulation strategies employed under variable operating conditions. The resulting system configuration is then integrated into the optimization and assessment framework developed in the previous chapters. Finally, the impact of the proposed technological transition on overall system performance is analyzed and compared against the previous configuration.

## 7.1 System configuration with radial turbomachinery

As stated in the chapter introduction, the CAES system configuration analyzed from this point onward considers the use of a centrifugal compressor and a radial turbine. A schematic representation of the system under investigation is provided in Figure 7.1.

The system analyzed is conceptually analogous to the configuration discussed in the previous chapter, consisting of a hybrid photovoltaic-CAES-diesel system coupled to an electrical load. Nevertheless, several modifications are introduced. In particular, radial turbomachinery is adopted for both the compression and expansion phases, each implemented through a multi-stage arrangement. The modeling and performance characteristics of these machines are discussed in detail in the following sections. The main plant-level differences with respect to the previously analyzed configuration, in addition to the use of radial machines, can be summarized as follows:

- **Number of stages.**

Both the compression and expansion processes are implemented through four stages. The choice to divide the compression/expansion in one more stage compared to the previous layout is motivated by the fact that centrifugal compressors and radial turbines typically operate with lower achievable pressure ratios per stage compared to reciprocating or other volumetric machines [161].

- **Thermal energy storage configuration**

Two separate, fully mixed, thermal energy storage units are adopted, operating at low and high temperature levels, respectively, instead of a single stratified storage. This choice avoids excessively increasing the height of the storage tank in order to preserve an acceptable aspect ratio at larger scales, which could otherwise lead to structural and mechanical issues.

- **Heat exchanger**

A further modification concerns the heat exchangers. Double-pipe

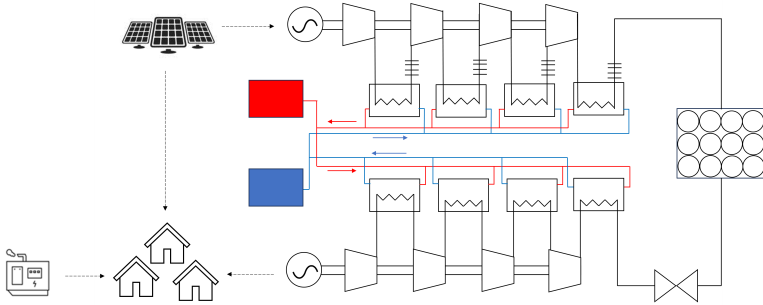


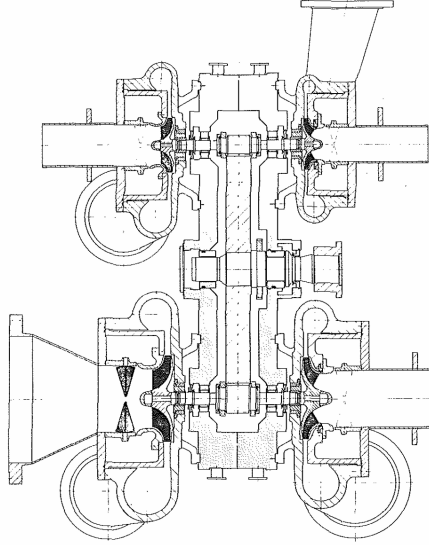
Figure 7.1: Plant configuration with the integration of a radial machine

heat exchangers are considered for heat transfer areas up to  $10 \text{ m}^2$ , whereas shell-and-tube heat exchangers are adopted beyond this threshold. From a modeling perspective, the same  $\varepsilon - NTU$  formulation is retained, with the differences between the two technologies accounted for through the definition of the heat exchanger effectiveness. In particular, shell-and-tube heat exchangers with two shell passes are considered in order to achieve an effectiveness close to that of an ideal counter-flow configuration, thereby enhancing thermal regeneration and mitigating the icing issues discussed previously.

### 7.1.1 Turbomachinery architecture and control strategies

The machines considered to be applied in such a CAES system are four-stage integrally geared machines. An integrally geared turbomachine is a multi-shaft configuration in which the individual stages are mechanically coupled through an internal gearbox. In the four-stage configuration considered here, the machine features a central low-speed shaft connected to the motor/generator, which drives two high-speed pinion shafts via the gearbox. The compression (or expansion) stages are arranged in pairs on the two high-speed shafts, with each shaft carrying two stages. This paired-stage arrangement allows a more balanced mechanical layout by

counteracting axial thrusts between coupled stages, which further justifies the adoption of a four-stage configuration. A cross-sectional view of a machine of this type is shown in Figure 7.2.



*Figure 7.2: Cross section of a four stage geared compressor [250]*

As for the system analysed in the previous chapters, the compressor is required to follow the time-varying power available from the photovoltaic generator, whereas the turbine should track, as closely as possible, the load demand of the end user. It is therefore necessary to consider appropriate power regulation strategies. In the previous chapters, variable-speed operation was considered as the sole regulation method, as it represents the primary approach for reciprocating and positive-displacement machines. For turbomachinery, however, a broader set of control options can be envisaged. In [123], an extensive review of control methods applied to turbomachinery is presented, with particular emphasis on their potential application within compressed air energy storage (CAES) systems. With specific reference to the compression phase, compressor off-design

control strategies in CAES applications can be classified into inlet/outlet valve control, rotating speed control, variable inlet guide vane (IGV) control, and variable diffuser vane control. The first approach modifies the inlet or outlet pressure and, consequently, the compressor mass flow rate; the second varies the rotational speed by means of an electric drive; the third and fourth approaches alter the compressor geometry, thereby modifying its characteristic curves. The study highlights that back-pressure or valve-based control strategies may lead to significant pressure losses. Variable diffuser vane control, on the other hand, exhibits limited capability in adjusting the pressure ratio, which is a critical requirement in CAES systems due to the large pressure variations occurring in the air storage reservoir. Conversely, rotating speed control and IGV control emerge as the most efficient and promising solutions, despite each presenting inherent drawbacks. In CAES applications, rotating speed control is associated with high costs due to the need for power electronic converters, particularly in large-scale systems. IGVs introduce additional mechanical complexity; however, they enable a wide operating range and high control efficiency.

In the present study, an inlet guide vane control strategy is adopted to modulate the compressor operating point while preserving acceptable operability margins. IGVs are assumed to be installed at the inlet of each of the four compression stages. This choice is motivated by the multi-stage, integrally geared architecture of the centrifugal compressor under consideration, for which a single variable-speed drive (VSD) would primarily affect all pinion shafts simultaneously and would not provide stage-wise degrees of freedom. As shown in [123], such an approach is not suitable for multi-shaft geared compressors. Conversely, independently actuated IGVs enable a distributed regulation of the individual stage characteristics, which is beneficial for maintaining high efficiency. The advantages of adopting IGV regulation in a four-stage integrally geared centrifugal compressor have been extensively demonstrated by Simon et al. [250]. Further details regarding the modelling and control implementation of this regulation strategy are provided in the following sections.



In an analogous way to the compression phase, power regulation during the expansion phase can be achieved through variable stator vane (VSV) control, which is conceptually similar to IGV regulation. In this case, the angular position of the stator vanes is adjusted, resulting in corresponding variations of the nozzle throat area. This, in turn, allows the regulation of key turbine parameters such as mass flow rate and specific enthalpy drop. Such a control strategy enables improved performance under off-design operating conditions and significantly extends the operational range of the turbine. Variable stator vane adjustment is a widely adopted and highly effective method for modifying the off-design behaviour of expansion machines and it represents a particularly attractive solution for CAES turbine applications [123].

Moreover, only a limited number of studies have investigated the performance of IGVs and VSVs control in CAES operation [39]. Historically, CAES applications were primarily aimed at peak shaving, a context that did not require a high degree of operational flexibility under off-design conditions. More recently, the increasing penetration of renewable energy sources has placed greater emphasis on the capability of CAES systems to operate efficiently over a wide range of operating conditions. This clearly motivates a deeper investigation into the off-design behaviour of CAES systems equipped with IGV/VSV-based control strategies.

## 7.2 Turbomachinery Modeling

The turbomachines are modeled under the assumption of adiabatic operation. This assumption is significantly more realistic for turbomachinery than for volumetric machines, as turbomachines typically exhibit much smaller heat transfer rates with the surroundings, compared to the machine power. The modeling approach adopted for turbomachinery relies on the use of performance maps, consistently with the methodology previously applied to reciprocating and volumetric machines. In accordance with classical turbomachinery literature, the characteristic parameters are expressed in terms of reduced (or corrected) quantities,

allowing the similarity theory to be exploited and enabling the extension of performance data over a wide operating range. The quantities considered are the pressure ratio ( $\beta$ ), reduced mass flow rate ( $M_r$ ), reduced rotational speed ( $N_r$ ), and isentropic efficiency ( $\eta_{is}$ ). Performance maps are normalized with respect to their design-point values. Henceforth, the ratio between a quantity and its corresponding design value is denoted by an overbar.

$$\bar{M}_r = \frac{M_r}{M_{r,n}}, \quad M_r = \frac{\dot{m}\sqrt{T_1}}{p_1} \quad (7.1)$$

$$\bar{N}_r = \frac{N_r}{N_{r,n}}, \quad N_r = \frac{N}{\sqrt{T_1}} \quad (7.2)$$

$$\bar{\beta} = \frac{\beta}{\beta_n} \quad (7.3)$$

$$\bar{\eta}_{is} = \frac{\eta_{is}}{\eta_{is,n}} \quad (7.4)$$

Subscripts  $n$  and 1 denote design-point values and inlet conditions, respectively.

### 7.2.1 Compressor

To describe the compressor operating characteristics, a generalized analytical formulation of the performance map is adopted. In particular, the approach proposed by Zhang and Cai [251] is employed, which is applicable to both centrifugal and axial-flow compressors. This formulation is grounded in physical considerations and calibrated using extensive experimental datasets.

The normalized pressure ratio is expressed as a quadratic function of the reduced mass flow rate:

$$\bar{\beta} = c_1 \bar{M}_r^2 + c_2 \bar{M}_r + c_3, \quad (7.5)$$

while the normalized isentropic efficiency is given by

$$\bar{\eta}_{is} = [1 - c_4(1 - \bar{N}_r)^2] \frac{\bar{N}_r}{\bar{M}_r} \left( 2 - \frac{\bar{N}_r}{\bar{M}_r} \right). \quad (7.6)$$

The coefficients  $c_1$ ,  $c_2$ , and  $c_3$  are functions of the reduced rotational speed and are defined as

$$c_1 = \frac{\bar{N}_r}{p(1 - q/\bar{N}_r) + \bar{N}_r(\bar{N}_r - q)^2} \quad (7.7)$$

$$c_2 = \frac{p - 2q\bar{N}_r^2}{p(1 - q/\bar{N}_r) + \bar{N}_r(\bar{N}_r - q)^2} \quad (7.8)$$

$$c_3 = -\frac{pq\bar{N}_r - m^2\bar{N}_r^3}{p(1 - q/\bar{N}_r) + \bar{N}_r(\bar{N}_r - q)^2} \quad (7.9)$$

$$c_4 = 0.3 \quad (7.10)$$

The parameters  $p$  and  $q$  depend on the compressor type. Values of  $p = 1.8$  and  $q = 1.8$  are typically adopted for centrifugal compressors, whereas  $p = 0.36$  and  $q = 1.06$  are used for axial-flow compressors. The coefficients  $p$  and  $q$  are obtained through regression of extensive experimental datasets reported in the literature [251]. Since a centrifugal compressor is considered in this chapter, both  $p$  and  $q$  are set equal to 1.8. A graphical representation of the resulting characteristic maps is reported in Figure 7.3.

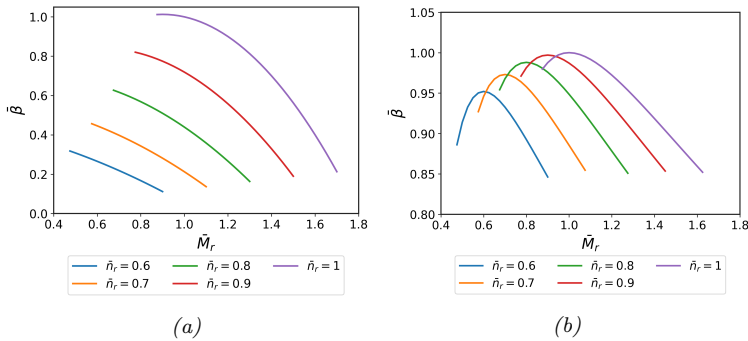


Figure 7.3: Compressor characteristics maps: (a) Pressure ratio, (b) Efficiency

The effect of variable inlet guide vanes on compressor performance is modeled using the correlations proposed by Haglind [252], which are widely employed in the evaluation of compressor characteristics under IGV regulation. The performance parameters at a generic IGV angle are obtained from the corresponding design-angle values by applying a transformation to the reference performance maps.

Let  $\Delta\alpha$  denote the IGV rotation angle, while the subscript map refers to quantities evaluated at the design IGV angle. The transformed parameters are computed as

$$M_r = M_{r,\text{map}} \left( 1 + \frac{b_1 \Delta\alpha}{100} \right) \quad (7.11)$$

$$\beta - 1 = (\beta_{\text{map}} - 1) \left( 1 + \frac{b_2 \Delta\alpha}{100} \right) \quad (7.12)$$

$$\eta_{is} = \eta_{is,\text{map}} \left( 1 - \frac{b_3 \Delta\alpha^2}{100} \right) \quad (7.13)$$

The coefficients  $b_1$ ,  $b_2$ , and  $b_3$  are set to 1, 1, and 0.01, respectively. The characteristic curves obtained by applying these transformations for different values of the IGV rotation angle are shown in Figure 7.4 for

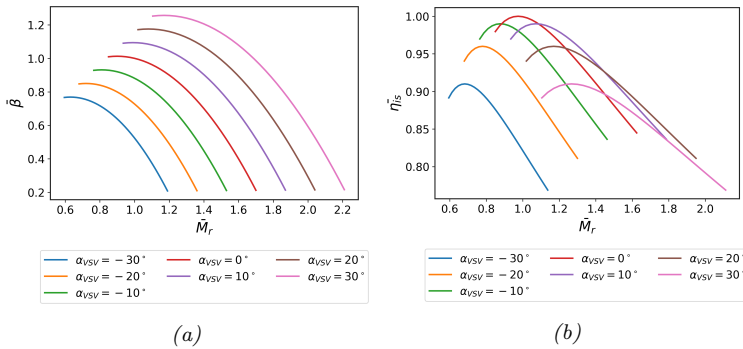


Figure 7.4: Compressor characteristics maps under different IGV angle and  $\bar{N}_r = 1$ : (a) Pressure ratio, (b) Efficiency

the case of  $\bar{N}_r = 1$ . The main effects of the inlet guide vanes (IGV) are illustrated in the plot. Closing the IGV (i.e., negative vane angles) leads to a reduction in the mass flow rate at a given pressure ratio. Moreover, variations in the IGV angle modify the flow incidence at the impeller inlet, resulting in a decrease in compressor efficiency.

### 7.2.2 Radial Turbine

Based on the same literature sources, an analytical representation of the characteristic curves of a radial turbine is also adopted. The turbine mass flow characteristics are described using a modified form of the Flügel equation:

$$\bar{M}_r = \sqrt{1.4 - 0.4 \frac{\bar{N}_t}{\bar{N}_{t,n}}} \sqrt{\frac{1/\beta^2 - 1}{1/\beta_n^2 - 1}} \quad (7.14)$$

while the normalized isentropic efficiency is given by

$$\bar{\eta}_{is} = [1 - t_4(1 - \bar{N}_r)^2] \left( \frac{\bar{N}_r}{\bar{M}_r} \right) \left( 2 - \frac{\bar{N}_r}{\bar{M}_r} \right) \quad (7.15)$$

General turbine maps according to Equations (7.14) and (7.15) are reported in Figure 7.5.

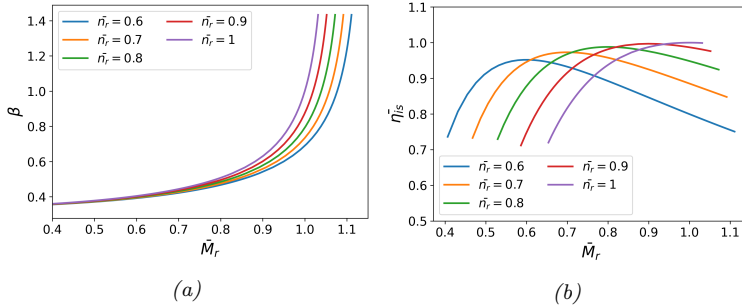


Figure 7.5: Turbine characteristics maps: (a) Pressure ratio, (b) Efficiency

Using the same notation previously introduced for the compressor, the

influence of turbine regulation through variable stator vanes is accounted for by applying the following transformations to the quantities derived from the characteristic maps in the reference case of zero VSV rotation angle.

$$\frac{M_r}{M_{r,\text{map}}} = 1.1875 - \frac{3}{10000}(\Delta\alpha - 25)^2 \quad (7.16)$$

$$\frac{\eta_{is}}{\eta_{is,\text{map}}} = \left(1 - \frac{0.9}{10000} \Delta\alpha^2\right). \quad (7.17)$$

An illustrative example of the application of these transformations is shown in Figure 7.6, where the transformations are applied for the case  $\tilde{N}_r = 1$ .

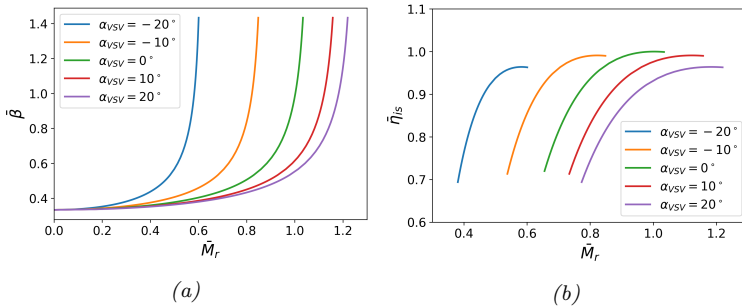


Figure 7.6: Turbine characteristics maps under different VSV angle and  $\tilde{N}_r = 1$ : (a) Pressure ratio, (b) Efficiency

### 7.2.3 Maximum efficiency estimation

The performance maps previously described are scaled according to design values of characteristic parameters. While the design mass flow rate and the compression/expansion ratio can be treated as adjustable design parameters (within a reasonable range dictated by the technological application) the same does not hold for the isentropic efficiency, which is inherently linked to the technological level and manufacturing quality of

the machines.

Under conventional operating conditions, a well-designed compression stage can typically achieve isentropic efficiencies in the range of 85–88%, with slightly higher values generally reported for turbine stages [253]. However, when this technology is scaled down to smaller sizes, several technological challenges arise. According to Euler’s turbomachinery equation, the amount of work transferred to or extracted from the working fluid is strongly dependent on maintaining sufficiently high blade peripheral velocities[254]. As the characteristic size of the machine decreases, achieving comparable peripheral speeds requires a significant increase in rotational speed, which is ultimately constrained by mechanical, structural, and bearing-related limits. Moreover, at reduced scales, clearance effects become dominant and represent a major source of performance degradation. Leakage losses associated with tip clearances scale can only be mitigated by increasing machining precision, at the expense of substantially higher manufacturing costs. These limitations are widely recognized in the literature as key factors limiting the efficiency of small-scale turbomachinery [254, 255].

The present work therefore aims to account for the expected efficiency penalties arising from the use of conventional turbomachinery technology at small scales since only commercially available, standard machines are considered in this analysis. For this reason, upper bounds on the allowable rotational speed are imposed, consistent with current industrial practice, as will be discussed in greater detail in the following section. The objective is to quantify the performance losses that arise when standard turbomachinery is employed in a reduced-scale context, and to assess whether a lower-scale limit exists beyond which alternative technologies, such as volumetric machines, become preferable for application in CAES systems.

To estimate the achievable efficiencies, the preliminary turbomachinery design methodology proposed by Balje is adopted [255, 256]. This approach is based on similarity theory and provides a consistent framework for scaling performance parameters and efficiency trends across different

machine sizes.

### 7.2.3.1 Theory background

In similarity analyses of turbomachinery, machine efficiency can be expressed as a function of the non-dimensional flow and load coefficients:

$$\eta = f(\Phi, \Psi). \quad (7.18)$$

Where the flow coefficient is defined as:

$$\Phi = \frac{c_x}{u_2}, \quad (7.19)$$

while the load coefficient is given by:

$$\Psi = \frac{\Delta h_{is}}{u_2^2}, \quad (7.20)$$

where  $c_x$  denotes the axial flow velocity,  $u_2$  is the peripheral blade speed and  $\Delta h_{is}$  represents the isentropic specific enthalpy change across the machine.

Although the coefficients  $\Phi$  and  $\Psi$  are widely adopted in aerodynamic analyses and detailed turbomachinery design, their direct use is less convenient during the early stages of system-level design. This limitation arises from their explicit dependence on geometric and kinematic quantities, which are generally unknown at the conceptual design stage. Moreover, since these coefficients are intrinsically defined for turbomachinery, they lack general applicability when different machine technologies are considered.

To overcome these limitations, similarity parameters based on specific speed and specific diameter, denoted as  $\omega_s$  and  $D_s$ , are more commonly employed for preliminary design and technology selection:

$$\omega_s = \omega \frac{\dot{V}^{1/2}}{w^{3/4}} = \frac{\Psi^{1/4}}{\Phi^{1/2}}, \quad (7.21)$$

$$D_s = D \frac{w^{1/4}}{\dot{V}^{1/2}} = \frac{\Phi^{1/2}}{\Psi^{3/4}}, \quad (7.22)$$



where  $D$  is a representative characteristic length of the machine, such as the rotor diameter for turbomachinery or the cylinder diameter for reciprocating devices. Since the specific speed and specific diameter can be expressed as combinations of the flow and load coefficients, the machine efficiency may equivalently be formulated as

$$\eta = f(\omega_s, D_s). \quad (7.23)$$

This representation is particularly advantageous for preliminary design and comparative assessment of turbomachinery technologies [257].

For a given specific speed, an optimal value of the specific diameter can be identified that maximizes machine efficiency:

$$D_{s,\text{opt}} = f(\omega_s). \quad (7.24)$$

The existence of such an optimal relationship was first highlighted by Cordier [258], who identified a statistical correlation between specific speed and the corresponding optimal specific diameter based on experimental data. This relationship, commonly referred to as *Cordier's line*, was originally developed for pumps. Subsequently, Baljé [255, 256] extended this concept by compiling efficiency maps for a wide range of driving and driven turbomachines, expressing efficiency as a function of specific speed and specific diameter. These statistical diagrams have since become a standard reference for turbomachinery selection and conceptual design. Figure 7.7 shows the Baljé diagram for both operating and driving machines, where the specific speed is expressed in dimensional form using units of the American measurement system, with the enthalpy rise represented as head in feet, as reported in the following equation:

$$N_s = \frac{N(\text{rpm}) \cdot \dot{V}(\text{ft}^3/\text{s})}{\Delta H_{is}(\text{ft})} \quad (7.25)$$

A generalized analytical formulation of Cordier's line can be derived [254]; the optimal specific diameter for pumps and compressors can be expressed as:

$$D_s = \begin{cases} 2.781 \omega_s^{-0.988}, & \text{for } \omega_s < 0.84, \\ 3.147 \omega_s^{-0.281}, & \text{for } \omega_s \geq 0.84, \end{cases} \quad (7.26)$$

while, for turbines, the optimal specific diameter is given by

$$D_s = 2.56 \omega_s^{-0.812}. \quad (7.27)$$

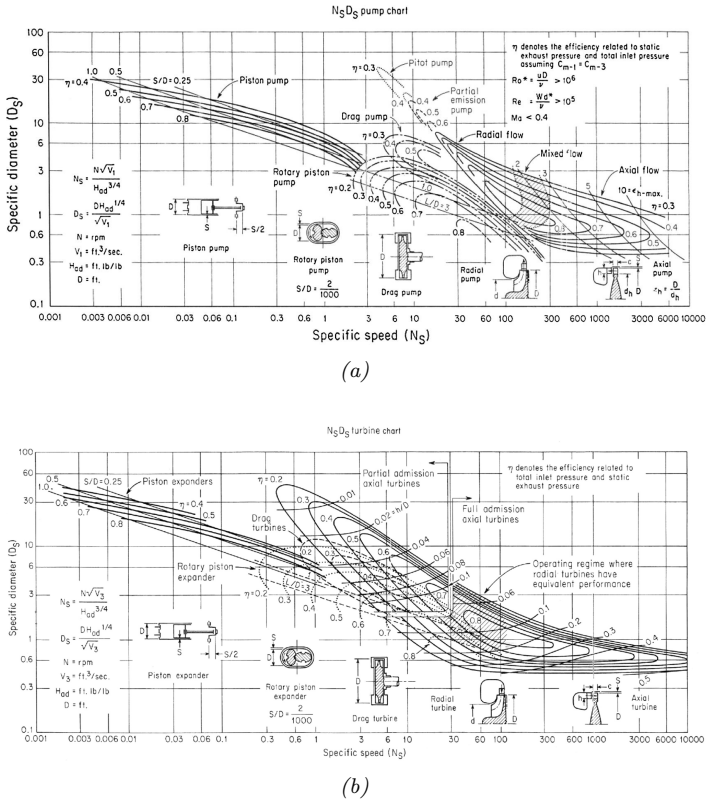


Figure 7.7: Baljé efficiency diagrams as a function of specific speed and specific diameter: (a) compressors and pumps, (b) turbines. from [255, 256]

### 7.2.3.2 Practical implementation

The application of preliminary turbomachinery design methodologies, as the Balje diagram, results in the estimation of relatively high rotational speeds when compact compressor or turbine configurations are pursued in order to achieve near optimal impeller peripheral velocities. Although the Balje approach enables operation within regimes where excessive tip speeds are not expected to be critical, high rotational speeds inherently introduce several challenges. These include limitations associated with maximum allowable peripheral velocity, rotor-dynamic stability, electric motor rotor integrity, and bearing load capacity. In addition, the use of electric machines capable of sustaining such elevated rotational speeds poses further mechanical and control-related constraints. Consequently, even when preliminary design tools indicate acceptable aerodynamic conditions, these aspects remain open and require careful evaluation.

Classical turbomachinery literature reports typical operating rotational speeds for centrifugal compressors in the range of 20–30 krpm [172]. However, in recent decades, technological advancements have significantly extended these limits, particularly for small and micro-scale applications. In this operating regime, achieving high efficiency frequently necessitates a substantial increase in rotational speed, primarily due to the reduced impeller diameters. Recent studies indicate that centrifugal compressors in the 40–500 kW power range operate at rotational speeds between 15 and 80 krpm, while microturbines of comparable power may reach speeds as high as 120 krpm [259, 260]. A key enabling technology for such systems is the adoption of high-speed electrical machines supported by active magnetic bearings. In these configurations, the compressor and turbine are typically directly coupled to the electrical machine, which must therefore be specifically designed to withstand high rotational speeds from both a mechanical and electromechanical perspective. This includes rotor-dynamic behavior, bearing design, and power regulation and control systems.

From the turbomachinery perspective, when operating at very high rotational speeds the bearing system represents a critical component: the

safe, reliable, and efficient operation of the entire installation depends largely on bearing performance [261].

The adoption of high-speed electrical machines and magnetic bearing systems inevitably leads to a significant increase in system complexity and capital cost. Within the application context considered in this thesis, characterized by off-grid operation and a strong emphasis on simplicity and reliability, minimizing capital investment costs represents a primary design objective, as highlighted by previous chapters results. For this reason, the use of magnetic bearing technology has been intentionally avoided. In fact, while magnetic bearings enable higher rotational speeds and eliminate mechanical contact losses, their implementation in a medium-power, integrally geared centrifugal compressor would substantially increase system cost, control complexity, and reliance on active electronic components, while offering only marginal benefits in terms of overall system efficiency. Furthermore, the multi-shaft architecture and off-design operating conditions typical of CAES applications tend to favor passive, oil-lubricated bearing solutions, which are generally more robust and industrially mature.

Defining a single, universal upper limit for rotational speed that avoids the need for such advanced technologies is not straightforward, as it depends on several interrelated factors that are strongly influenced by machine size. These include rated power, impeller diameter, allowable peripheral velocity, and drivetrain configuration.

In the present thesis, a maximum rotational speed constraint of 50 krpm is imposed on the high-speed pinions during the system sizing process. This limit is based on the availability of industrially proven integrally geared centrifugal compressors equipped with standard oil-lubricated tilted-pad bearings [262, 263]. This design choice places the proposed system, at the upper boundary of conventional industrial technology, without resorting to non-standard or experimental solutions. Comparable operating ranges are reported in the literature. For instance, [130] presents the design of a microturbine for energy recovery from waste heat or compressed air, employing oil-film bearings and operating within a

rotational speed range of 15–50 krpm. Similarly, [264] describes a 120 kW micro gas turbine supported by air foil bearings, with a rated rotational speed of 45,800 rpm and a maximum overspeed of 50,400 rpm.

The stage efficiency is estimated through a preliminary similarity-based procedure, in which the effective rotational speed is first determined and subsequently used to identify the optimal similarity parameters and the corresponding peak efficiency. The procedure is structured as follows:

1. Starting from the design mass flow rate, the corresponding volumetric flow rate is computed, while the design pressure ratio is used to determine the associated isentropic specific work.
2. A target value of the specific speed,  $\omega_{s,\text{target}}$ , is selected according to Baljé-type turbomachinery selection criteria for the first and third compression/expansion stages (i.e. 0.45 for the turbine and 0.8 for the compressor).
3. Based on the selected  $\omega_{s,\text{target}}$ , the corresponding target rotational speed,  $N_{\text{target}}$ , is computed using the similarity definitions.
4. The computed value of  $N_{\text{target}}$  is compared with the maximum admissible rotational speed,  $N_{\text{max}}$ . If  $N_{\text{target}} \leq N_{\text{max}}$ , the effective rotational speed is set equal to  $N_{\text{target}}$ ; otherwise, it is limited to  $N_{\text{max}}$ .
5. Once the effective rotational speed  $N$  is defined, the actual specific speed  $\omega_s$  is computed according to its definition (Equation (7.21)).
6. The optimum specific diameter is then obtained from the Cordier correlation,  $D_{s,\text{opt}} = f(\omega_s)$ , providing an estimate of the optimal size parameter for the selected similarity level.
7. Finally, the expected peak efficiency is estimated as a function of the specific speed  $\omega_s$ , assuming  $D_s = D_{s,\text{opt}}$ , yielding a preliminary estimate of the maximum achievable stage efficiency for the selected machine size.

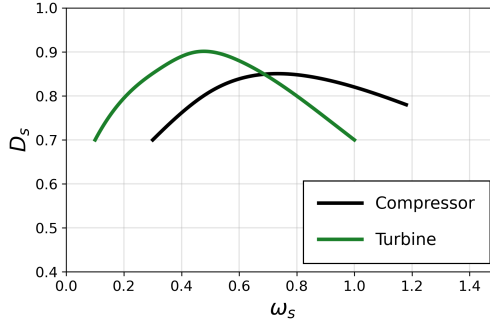


Figure 7.8: Maximum efficiency as a function of specific speed along the Cordier line

The maximum achievable efficiency along the Cordier line, expressed as a function of the specific speed, is reported for a centrifugal compressor stage and a radial turbine stage in Figure 7.8. The curves are obtained using the data reported in [254, 265].

These efficiency values must be further corrected to account for the effect of the Reynolds number. Since the work of Balje, it has been noted that efficiency values derived from similarity diagrams should be corrected to account for the actual Reynolds number of the machine [255]. This effect becomes particularly significant in cases where the Reynolds number is relatively low, as typically occurs in small-scale turbomachinery. Numerous correlations accounting for Reynolds number effects have been proposed in the turbomachinery literature. These correlations are generally expressed in the following form:

$$\frac{1 - \eta}{1 - \eta_{\text{ref}}} = a + (1 - a) \left( \frac{\text{Re}_{\text{ref}}}{\text{Re}} \right)^n \quad (7.28)$$

where  $\text{Re}$  denotes the Reynolds number, while  $a$  and  $n$  are empirical coefficients. A comprehensive review of the main correlations available in the literature is provided by [266], who shows that improved accuracy is generally obtained when the exponent  $n$  is itself expressed as a function of the Reynolds number.

In the present thesis work, the correlation proposed by Capata and Sciubba [267] is adopted, as it has been specifically developed for small-scale radial turbomachines. In this formulation, reported in Equation (7.29), the Reynolds number is defined as a function of the impeller diameter, which can be readily derived from the selected specific diameter. This feature makes the correlation particularly suitable for preliminary design stages, where the detailed geometry of the flow passages is not yet available.

$$\frac{1 - \eta}{1 - \eta_{\text{ref}}} = 0.50 + 0.50 \left( \frac{\text{Re}_{\text{ref}}}{\text{Re}} \right)^{0.084 \left( \frac{\text{Re}_{\text{ref}}}{\text{Re}} \right)^{0.25}} \quad (7.29)$$

with the Reynolds number defined as:

$$\text{Re} = \frac{uD}{\nu}, \quad (7.30)$$

where  $u$  is the peripheral velocity,  $D$  is the impeller diameter, and  $\nu$  is the kinematic viscosity of the working fluid.

At this stage, the degradation of efficiency associated with increased clearance-related losses is not explicitly accounted for. A reliable quantification of such effects would require more detailed information on the machine geometry.

The overall procedure is summarized by the flowchart shown in Figure 7.9.

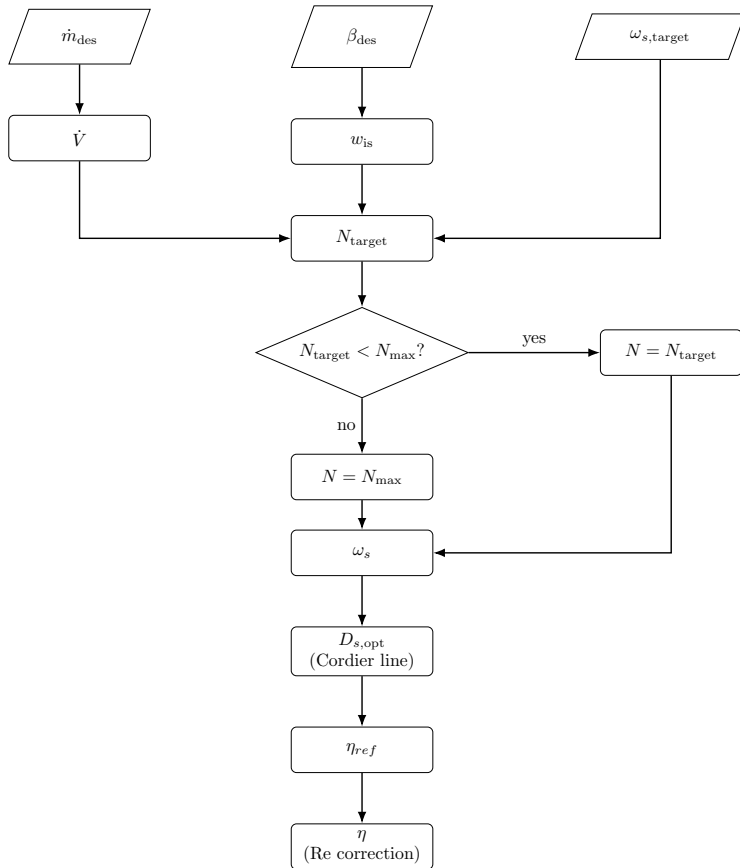


Figure 7.9: Flowchart for selecting the rotational speed and estimating the peak efficiency



### 7.3 System integration with renewable generation and load

As for the system analysed previously in Section 4.2.5, regulation is required during both the charging and discharging phases in order to track the instantaneous power imbalance between photovoltaic generation and load demand, defined as:

$$\Delta P(t) = P_{PV}(t) - P_{Load}(t), \quad (7.31)$$

while respecting the constraints imposed by the maximum allowable pressure in the storage vessel and by the characteristic performance curves of the turbomachinery.

As discussed in the previous sections, the control strategy implemented in this system relies on the use of inlet guide vanes and variable stator vanes. This approach differs from the case of volumetric machines, where regulation was achieved by varying the rotational speed. In the present configuration, regulation is performed through a proportional–integral (PI) controller [268], which modulates the opening angle of the first stage in order to match the required power by varying the mass flow rate. The vane angles of the stages downstream of the first one are instead prescribed according to the optimal values obtained from a prior optimization procedure, aimed to maximize the overall compression/expansion efficiency, described in the following sections. The remaining IGV/VSV angles  $\alpha_i$  ( $i = 2, \dots, 4$ ) are therefore scheduled as functions of  $\alpha_1$ . Within the PI control scheme, the setpoint corresponds to the power to be delivered or absorbed, while the measured variable is the instantaneous power absorbed or generated by the machine. The controller output adjusts the vane angle  $\alpha$  of the first compression or expansion stage within the allowable range ( $-30^\circ$  to  $30^\circ$  for the compressor and  $-25^\circ$  to  $25^\circ$  for the turbine).

The possibility of integrating of a PI control system within the thermodynamic model represents one of the key advantages of the Modelica language, which enables the unified modelling of multidisciplinary sys-

tems. The controller is implemented using components from the Modelica Standard Library [269].

## 7.4 IGV and VSV Angle Optimization

In order to generate maps providing the values of the IGV/VSV angles  $\alpha_i$  ( $i = 2, \dots, 4$ ) as functions of the first-stage angle  $\alpha_1$  and of the outlet/inlet pressure, the optimization procedure described below is adopted. For the sake of conciseness, the methodology is presented in detail for the compression train only; an analogous procedure is applied to the expansion phase.

### 7.4.1 Numerical method

With reference to the compression process, the optimization is based on a model of the compression train only. A PI controller is implemented to regulate the first IGV angle  $\alpha_1$  in order to target a prescribed value of reduced mass flow rate. The remaining three IGV angles are treated as independent decision variables and can be varied arbitrarily within their admissible ranges. A graphical representation of the model used during the optimization process is shown in Figure 7.10. As illustrated, a heat-sink is placed between each compression stage, restoring the air temperature to ambient conditions. This represents both the heat recovery and the subsequent heat rejection to the ambient present in the CAES system. In the present analysis, however, the focus is exclusively on the compression process, and these thermal subsystems are not explicitly modeled in detail. The inlet conditions are assumed to correspond to standard ambient conditions ( $p = 1$  bar,  $T = 25^\circ\text{C}$ ).

The model also includes the option to bypass one or two last stages, depending on the back pressure, whenever operating with all the inter-cooled stages is not beneficial compared to using fewer stages. Although multistage intercooled compression is generally advantageous, very low pressure ratios in the final stages may penalize compressor efficiency; in such cases, bypassing the last stages can yield a higher overall compression

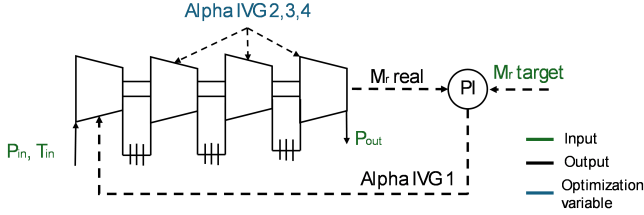


Figure 7.10: Schematic diagram of the IGV optimization strategy for the compression train

efficiency. The model is employed to perform quasi-steady-state simulations. A fixed value of back pressure and a fixed reduced mass flow rate are assumed, with no time variation of these quantities. To ensure that the PI controller reaches steady-state conditions and that transient effects are fully damped, each simulation is run for a total duration of 100 s, and only the final values are retained for the optimization process. It should be noted that, at this stage, the set-point and target variables of the PI controller are the reduced mass flow rates, whereas in the full system model the control strategy is based on the absorbed or delivered power.

All quantities are expressed in non-dimensional form, scaled with respect to their design values, in order to exploit similarity theory and obtain generalized results (i.e.  $\bar{M}_r = M_r/M_{r,n}$ ,  $\bar{P}_{out} = P_{out}/P_{out,n}$ ) allowing the resulting maps to be applied to different design mass flow rates and pressure levels during subsequent system optimization and scaling procedures.

To determine the optimal IGV angles, a genetic algorithm (GA) is employed. For each combination of outlet pressure and imposed reduced mass flow rate, the three downstream IGV angles are varied independently, while the first angle is controlled by the PI loop. This procedure is repeated over a sufficiently wide range of mass flow rates and pressures to cover the entire operating envelope of the compression system. The resulting data are used to construct maps that relate the optimal IGV angles of the downstream stages to the value of the first-stage angle. These maps are

subsequently implemented within the full CAES system model through the use of look-up table.

The objective function of the optimization is the specific work of the compression process. By minimizing the specific work, the combination of the four IGV angles that maximizes the compression efficiency is identified for each operating condition.

The model is developed in Modelica and, given its relatively simple structure, it can be efficiently exported via the Functional Mock-up Interface (FMI) standard [270]. The resulting executable model enables efficient co-simulation with external engineering software. In the present work, the FMI model is embedded within a Python environment, where the genetic algorithms provided by the DEAP library [271] are employed. This approach significantly reduces computational time and makes the procedure well suited for GA-based optimization, which inherently requires a large number of model evaluations. The main settings of the genetic algorithm are summarized in Table 7.1

It is worth noting that an analogous approach cannot be readily applied to the optimization of the entire CAES system model. The full model involves a large number of discrete events and state changes during simulation, which are difficult to handle robustly within an FMI-based co-simulation framework.

*Table 7.1: Genetic Algorithm parameters adopted for IGV/VSV angles optimization*

| Parameter             | Value     |
|-----------------------|-----------|
| Population size       | 200       |
| Number of generations | 60        |
| Crossover probability | 0.8       |
| Mutation probability  | 0.1       |
| Convergence tolerance | $10^{-5}$ |

As anticipated, an analogous procedure is adopted for the expansion phase. In this case, the pressure variable of interest is the turbine inlet pressure rather than the outlet pressure, and the objective function

is modified accordingly, with the aim of maximizing the specific work extracted from the working fluid rather than minimizing the absorbed work.

### 7.4.2 Results

This section presents the results of the optimization procedure described previously, organized into two subsections addressing the compressor and the turbine, respectively. All curves are reported in nondimensional form, with pressure levels and reduced mass flow rates normalized by their corresponding design values. This representation ensures applicability across different operating conditions while preserving machine similarity.

#### • Compressor

Figure 7.11 presents the results of the optimization process in terms of the first-stage IGV angle,  $\alpha_1$ , as a function of the normalized reduced mass flow rate of the first stage, for six different values of back pressure. The corresponding optimized values of the downstream IGV angles are also reported as functions of  $\alpha_1$ .

For the lowest back-pressure condition, only the optimized value of  $\alpha_2$  is available. At the second lowest back-pressure level, optimized values are obtained for both  $\alpha_2$  and  $\alpha_3$ , whereas for the remaining four cases all three downstream IGV angles are present. This behavior is a direct consequence of the integrated bypass strategy, which allows the exclusion of the final stages from operation under low back-pressure conditions.

The variation of the first-stage IGV angle with the target reduced mass flow rate,  $\bar{M}_r$ , for different back-pressure values is shown in Figure 7.11-(a). A nearly linear relationship between the required mass flow rate and the IGV rotation angle can be observed. This behavior arises from the modulation of the effective flow area under fixed inlet density conditions (ambient conditions for the first stage).

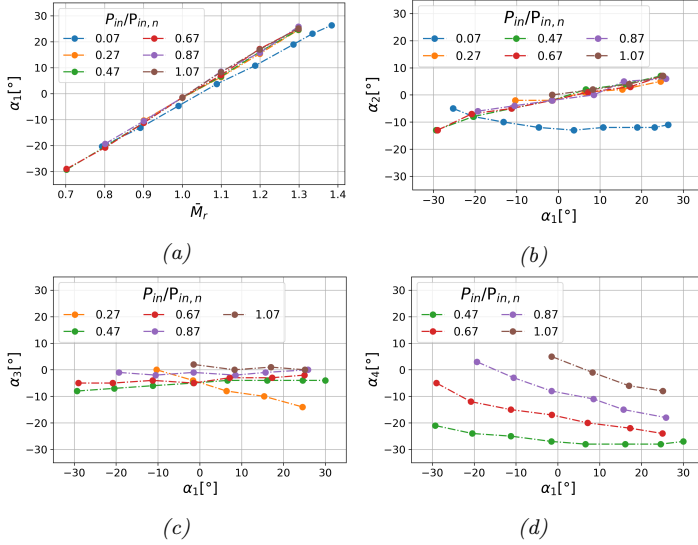


Figure 7.11: Regulation of IGV angles for different back pressure levels

The observed trend reflects the linear nature of the governing relation expressed in Equation (7.11).

The variations of the second, third, and fourth stage IGV rotation angles as functions of  $\alpha_1$  are reported in Figure 7.11-(b), (c), and (d), respectively. Compared with the first stage, the variation ranges of the downstream IGV angles with respect to the reduced mass flow rate are significantly smaller, with the 3<sup>rd</sup> stage IGV exhibiting the smallest variation. Overall, the IGV angles of the first three stages increase with increasing  $\bar{M}_r$ , although the rate of increase progressively decreases from the first to the third stage. In contrast, the fourth-stage IGV angle exhibits an opposite trend. Moreover, the fourth-stage angle shows a markedly stronger dependence on the total compression ratio than the upstream stages. It should be noted that for outlet pressure levels of  $0.07 P_{out,n}$  and  $0.27 P_{out,n}$ , the trends of  $\alpha_2$  and  $\alpha_3$  resemble those of the fourth stage in the

other operating conditions. This occurs because, in these cases, the second and third stages effectively act as the final active compression stage, as the subsequent stages are bypassed.

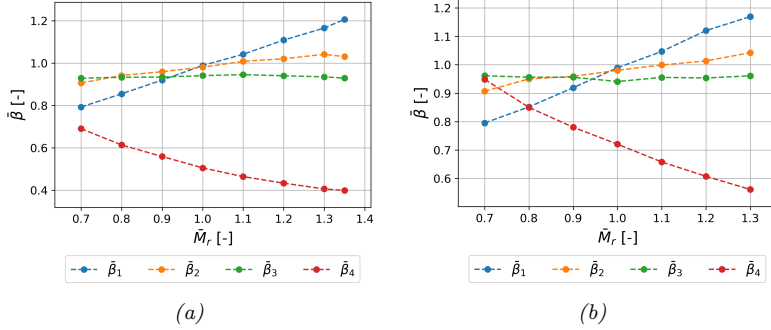


Figure 7.12: Normalized pressure ratio vs. normalized reduced mass flow rate for four stages: (a)  $P_{out}/P_{out,n} = 0.47$ , (b)  $P_{out}/P_{out,n} = 0.67$ .

The reasons for these behaviors can be understood through an analysis of the pressure ratio distribution among the stages. In brief, the IGV adjustments of the first three stages aim to maintain their operation close to their respective design points, thereby maximizing their isentropic efficiencies. In contrast, the fourth stage primarily serves to satisfy the required back-pressure constraint. Consequently, the optimization strategy prioritizes improving the isentropic efficiencies of the upstream stages, keeping them as close as possible to nominal operating conditions. Accordingly, the last stage is the one operating furthest from its optimal design point and therefore exhibits the lowest efficiency. Operating under off-design conditions, it is also characterized by a lower pressure coefficient,  $\beta$ , and consequently absorbs less work. This behavior is illustrated in Figure 7.12, which reports the normalized pressure coefficient,  $\tilde{\beta}$ , for the four compression stages at two different outlet pressure levels as a function of the first-stage reduced mass flow rate,  $\tilde{M}_{r,1}$ . As the mass flow rate increases, the pressure coefficient of the first

stage increases accordingly. In contrast, the pressure coefficients of the second and third stages remain nearly constant at values close to unity, indicating operation near their design conditions. The fourth stage exhibits an opposite trend compared with the first stage, with  $\bar{\beta}$  values progressively departing from unity, confirming that it operates further from its design point than the upstream stages.

#### • Turbine

The results obtained from the optimization procedure applied to the turbine are reported in Figures 7.13 and 7.14. Specifically, Figure 7.13 presents the results corresponding to pressure levels for which all four expansion stages are active, whereas Figure 7.14 shows the results obtained under operating conditions in which the first expansion stage is bypassed. Under these conditions, the mass flow rate is controlled by acting on the second stage VSV angle. Consequently, the optimal values of the downstream vane angles are expressed as functions of  $\alpha_2$  rather than  $\alpha_1$ .

Figure 7.13-(a) illustrates the variation of the first-stage VSV rotation angle as a function of the reduced mass flow rate,  $\bar{M}_r$ , and the inlet pressure. The results indicate that the first-stage VSV angle increases with increasing reduced mass flow rate, as a wider VSV opening is required to sustain the prescribed expansion ratio at higher mass flow conditions. Unlike compressors, in which the mass flow rate varies approximately linearly with the inlet guide vane rotation angle, the relationship between mass flow rate and VSV angle in turbines follows a quadratic trend, as expressed by Equation (7.16). This nonlinear behavior is clearly reflected in the shape of the curves reported in the figure. Figure 7.13-(b), (c), and (d) show the variation of the VSV rotation angles of the second, third, and fourth stages as functions of the first-stage VSV angle. It can be observed that the VSV angles of the downstream stages initially increase with that of the first stage and then tend to stabilize



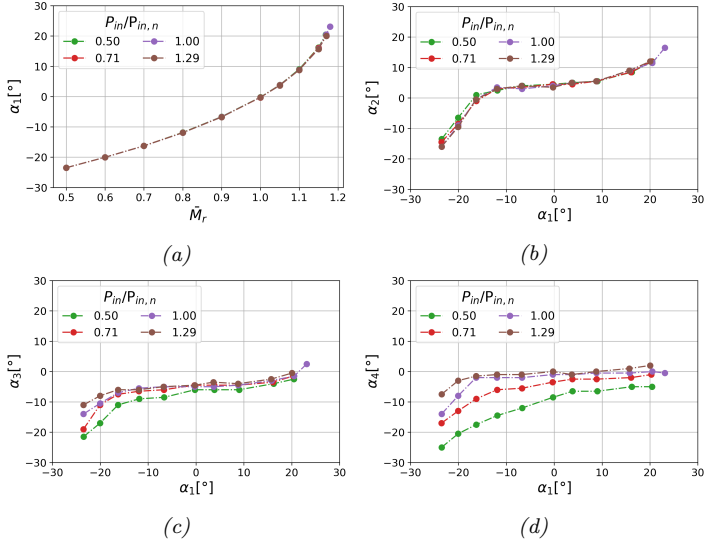


Figure 7.13: Regulation of VSV angles for different inlet pressure levels

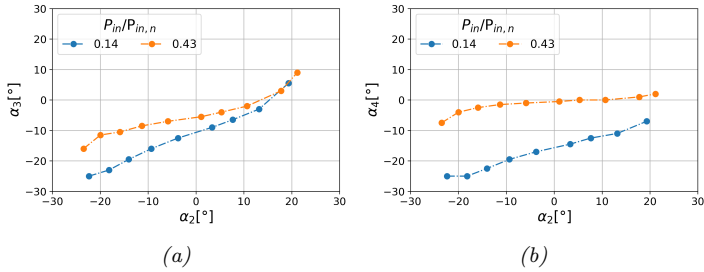


Figure 7.14: Regulation of VSV angles for different inlet pressure levels with the bypass of first expansion stage

around values close to 0°. The influence of inlet pressure on these angles is generally limited, although it becomes more pronounced for the last expansion stage.

As in the case of the compressor, additional insight into these trends

can be obtained by analyzing the distribution of the expansion ratio among the different stages, reported in Figure 7.15. Figure 7.15-(a) shows the expansion ratio distribution for an inlet pressure equal to 0.5 times the design value.

The first expansion stage consistently exhibits the highest expansion ratio, making it the stage that delivers the largest contribution to the overall turbine work. In addition, the first stage shows the most pronounced sensitivity to variations in the VSV angle. Its expansion ratio remains nearly constant at a value of approximately 1.4 for low VSV angles and then progressively decreases as the VSV angle increases. This behavior is consistent with the efficiency characteristics of the stage under variable VSV operation. As the VSV angle increases, the value of the expansion ratio corresponding to maximum efficiency shifts towards lower values. Therefore, in order to operate close to its optimal efficiency point, the expansion ratio of the first stage must be reduced accordingly. Moreover, expansion ratios higher than about 1.4 lead to choking conditions, which should not be exceeded. For this reason, in this operating region the VSV angles of the downstream stages are adjusted so as to prevent a further increase in the first-stage expansion ratio and to keep it close to its optimal value. This control strategy is ultimately driven by the need to optimize the operating conditions of the first stage, which delivers the largest share of the turbine work.

A different behavior is observed at higher inlet pressures. Under these conditions, as shown in Figure 7.15-(b), the fourth stage exhibits a significant increase in its expansion ratio, eventually becoming the stage characterized by the highest expansion ratio. This redistribution of the expansion ratios among the stages is required in order to keep the fourth-stage VSV angle close to zero. Given the shape of the efficiency curve, operating at VSV angles far from zero has a more pronounced negative impact on efficiency than operating at expansion ratios that deviate from their optimal value.

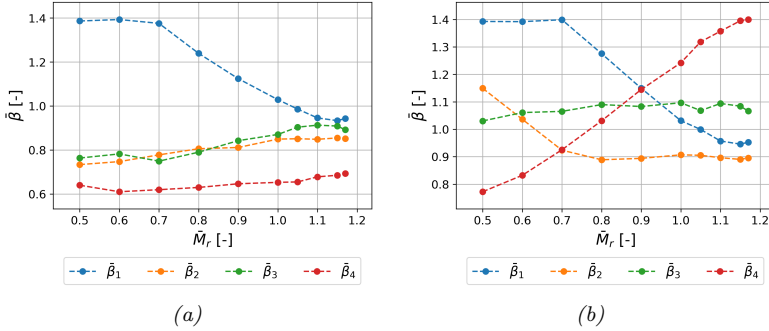


Figure 7.15: Normalized pressure ratio vs. normalized reduced mass flow rate for four stages: (a)  $P_{in}/P_{in,n} = 0.5$ , (b)  $P_{in}/P_{in,n} = 1.28$

For this reason, since the last stage delivers a significant fraction of the total turbine work, it is preferable to maintain its VSV angle close to zero, even at the expense of a non-optimal expansion ratio.

## 7.5 Case Study in hybrid off-grid configuration

As stated at the beginning of the chapter, one of its main objectives is to reproduce an analysis analogous to that carried out in the previous chapter, this time considering the new configuration of the CAES system. This approach ultimately enables a comparison between the two different system configurations. In particular, the present analysis focuses exclusively on Scenario 3 of the previous chapter, that is, the operating condition under which the hybrid CAES–PV–diesel system is considered. This configuration corresponds to the plant layout shown in Figure 7.1, presented at the beginning of the chapter.

In the following, the boundary conditions of the system and other information concerning the setup of the analyses are briefly recalled. Furthermore, a number of modifications with respect to the approach adopted in the previous chapter are introduced, which are required to account for the differences introduced by the new layout, particularly with respect to cost estimation and the definition of the optimization variables.

### 7.5.1 Economic analysis

The economic assessment is based on the evaluation of the LCOE of the overall system, accounting for both the electricity generated and the costs associated with all three energy sources (i.e. the PV system, the CAES system, and the diesel generator), as introduced in Section 6.2.2. In order to evaluate the LCOE for the new system configuration, it is necessary to introduce an estimation of the investment costs of those components that differ with respect to the configuration previously analyzed.

- **Turbomachinery**

The capital cost of the compressor and the turbine is expressed as a function of their design power, according to the correlations proposed in [272] and [182], respectively:

$$CAPEX_{\text{compressor}} = 8100 \cdot P^{0.6} \quad (7.32)$$

$$CAPEX_{\text{turbine}} = 10^{(2.2476 + 1.4965 \log_{10}(P) - 0.1618 [\log_{10}(P)]^2)} \quad (7.33)$$

- **Shell-and-Tube Heat Exchangers**

The capital cost of shell-and-tube heat exchangers is estimated using the correlation proposed by Din et al. [273], expressed as a function of the heat transfer area:

$$CAPEX_{\text{HEX}} = 8500 + 409 \cdot A^{0.85} \quad (7.34)$$

### 7.5.2 User load demand and system location

As described in Section 6.2.5, the system supplies an end user characterized by a load profile representative of a typical domestic demand. This load profile is recalled for reference in Figure 7.16. The load demand is parameterized and scaled in order to perform a parametric analysis

with respect to the size of the end user, which is here defined in terms of the total energy demand. The analyzed range spans from a minimum case corresponding to an energy demand of 1 MWh/day, with a peak power of approximately 50 kW, up to a system size ten times larger.

Seven installation sites are considered, namely the same locations analyzed in the previous chapters (i.e., Romania, Cambodia, India, Belize, Jordan, Botswana, and Zimbabwe). Their characteristics in terms of photovoltaic potential and diesel fuel cost are summarized in Table 6.2 in Section 6.2.5. As will be discussed in more detail in the results in Section 7.6, the parametric analysis is carried out only for Zimbabwe, which was identified in the previous chapter as the most promising location. All the remaining locations are instead analyzed only at the maximum system size.

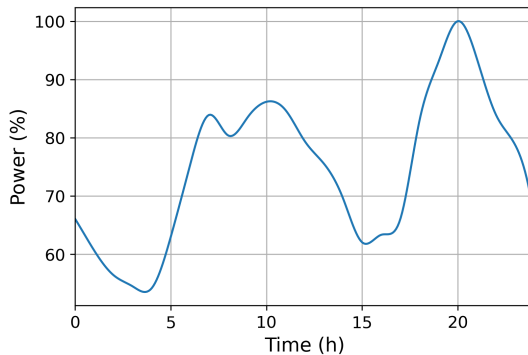


Figure 7.16: End user power request

### 7.5.3 Sizing optimization

As in the previous chapters, a sizing procedure based on an optimization process is implemented in order to determine the optimal sizes of the various system components. The optimization procedure adopted is the same as that used in the previous analysis and is based on the application

of Efficient Global Optimization algorithms. The objective function to be minimized is the levelized cost of electricity of the overall system.

With regard to the newly introduced compressor and turbine, it is necessary to modify the scaling parameter with respect to the previous case, in which the machines were scaled based on the swept volume. In the present configuration, the machines are instead scaled on the basis of their design mass flow rate. For each optimization iteration, the corresponding design-point values used to scale the performance maps are derived by assuming that the design pressure ratio is equally distributed among all compression and expansion stages. For the turbine, a maximum inlet pressure of 100 bar is assumed, in agreement with the current state of the art of small-scale expanders [274]. This limit is enforced by means of a throttling valve placed upstream of the turbine. Based on the design mass flow rate and the design pressure, the reduced mass flow rate used to scale the performance curves is determined. The design efficiency is then derived following the procedure described in the previous sections.

The parameters characterizing the remaining components (i.e. the pressure storage vessels and the PV system) are kept unchanged. The parameters investigated during the optimization process are summarized in Table 7.2, together with their corresponding exploration ranges. It should be noted that the ranges reported refer to the minimum system size considered in the parametric analysis discussed in the previous section; for the other cases, the corresponding bounds are subsequently scaled in accordance with the overall system size scaling.

*Table 7.2: Decision variables and bounds for the turbomachinery-based system sizing optimization (minimum-scale case)*

| Parameter                        | Symbol          | Range                        |
|----------------------------------|-----------------|------------------------------|
| Compressor design mass flow rate | $\dot{m}_{c,n}$ | $0.1 \div 1 \text{ kg/s}$    |
| Turbine design mass flow rate    | $\dot{m}_{t,n}$ | $0.1 \div 1 \text{ kg/s}$    |
| Pressure vessels volume          | $V$             | $10 \div 200 \text{ m}^3$    |
| Max air pressure inside vessels  | $p_{max}$       | $40 \div 200 \text{ bar}$    |
| PV installed peak power          | $P_{PV,p}$      | $100 \div 1000 \text{ kW}_p$ |

## 7.6 Results

The results obtained for different system sizes at the Zimbabwe location are presented in Figure 7.17, where the Levelized Cost of Energy is shown as a function of system size, expressed in terms of the daily energy demand of the end user. An approximately hyperbolic trend of the LCOE as a function of system size can be observed. The maximum LCOE value, equal to 0.29 \$/kWh, corresponds to a system configuration characterized by a daily energy demand of 1 MWh/day. As the energy demand increases, the LCOE decreases rapidly up to about 3 MWh/day. Beyond this threshold, the rate of decrease becomes more gradual, reaching a value of 0.202 \$/kWh for a system with a daily energy demand of 10 MWh/day.

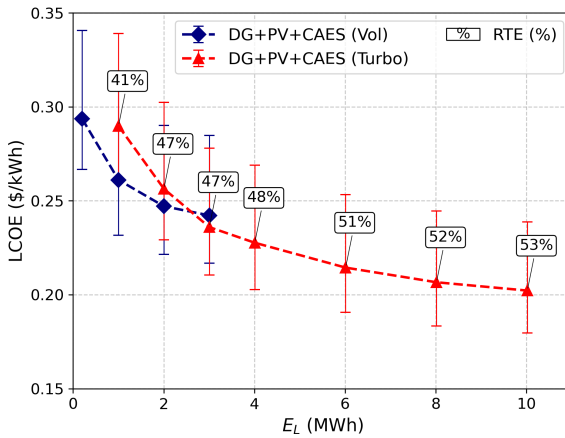


Figure 7.17: LCOE of the hybrid system at different system scale

A comparison with the results previously obtained for the configuration based on positive displacement machines, shown in Figure 6.16 and reproduced here for reference, indicates that the two LCOE curves intersect at a daily energy demand of approximately 3 MWh. Below this value, the configuration employing volumetric machines results in lower LCOE values, whereas above this threshold the turbomachinery-based

configuration becomes economically preferable. This behavior reflects a clear technological trade-off, with the intersection point identifying the energy demand at which the economic advantage shifts between the two technologies. The intersection corresponds to a CAES system characterized by a compressor rated at approximately 400 kW, a turbine rated at about 150 kW, and an overall storage capacity of roughly 1.4 MWh.

The plot in Figure 7.17 also reports the round-trip efficiency associated with the various optimized system configurations. This result is particularly noteworthy because, unlike the case with positive displacement machines, where the round-trip efficiency remained nearly constant at approximately 35%, the turbomachinery-based configuration exhibits a marked increase in efficiency with increasing system size. Specifically, the RTE rises from values of about 41% for smaller systems to approximately 54% for the largest configurations considered. For smaller-scale systems, however, the higher efficiency achieved with turbomachinery is not sufficient to offset the increased costs associated with the adoption of a more technologically complex solution. As a result, despite the efficiency advantage, the overall economic performance remains less favorable compared to the configuration employing volumetric machines.

The observed variation in round-trip efficiency at the system level is primarily attributable to changes in the isentropic efficiency of the machines as a function of their size. As discussed previously in Section 7.2.3, the model incorporates a procedure to estimate the maximum achievable isentropic efficiency while accounting for efficiency degradation associated with downscaling the machines. This limitation arises from the constraint imposed on the rotational speed, which is not allowed to exceed a prescribed upper limit of 50 krpm. (This assumption is introduced to represent standard turbomachinery technologies and to avoid transitioning toward miniaturized, ultra-high-speed machines, which would imply substantially increased costs.)

This phenomenon can be observed in Figure 7.18, where the efficiencies of the four compression and expansion stages are reported for the various optimized cases as a function of machine power.



For the compression train, the stage efficiencies progressively decrease from the first to the fourth stage. This behavior is primarily due to the reduction in volumetric flow rate along the stages, which leads to a decrease in the specific speed  $\omega_s$ , causing it to deviate from its optimal value. For all stages, the efficiency generally increases with increasing machine size. This trend can be explained by the fact that, while maintaining the rotational speed fixed at the imposed upper limit of 50 krpm, an increase in volumetric flow rate shifts the operating point toward more favorable values of  $\omega_s$ , thereby improving efficiency. Beyond a certain machine size, however, the efficiency of the first and second stages no longer increases. In these configurations, the first stage is able to reach the target value of  $\omega_s$  (i.e., 0.8) at rotational speeds lower than 50 krpm. As the machine size increases further, the required rotational speed decreases accordingly, while the efficiency remains unchanged due to the constant value of  $\omega_s$ . This behavior does not apply to the third and fourth stages, as the shaft on which they are mounted would need to rotate at even higher speeds to achieve the same specific speed values. This limitation arises from the lower volumetric flow rates associated with the increased pressure levels reached through compression.

Analogous considerations apply to the turbine stages, as shown in Figure 7.18-(b). When the shaft rotational speed is constrained to the imposed upper limit, the downstream stages of the expansion process generally exhibit higher efficiency, owing to the larger associated volumetric flow rates. Once the optimal specific speed can be achieved on the first stage of a given shaft, this stage exhibits the highest efficiency, while the subsequent stage operates slightly away from its optimal conditions. This behavior is clearly observed for the shaft carrying the third and fourth expansion stages, as illustrated in the figure.

The main design parameters and performance indicators of the optimized configurations corresponding to the different system sizes are summarized in Table 7.3. An analysis of the pressure levels shows that, for small system sizes (and correspondingly small machines), the optimal solution is obtained at relatively low pressure values. As the system size

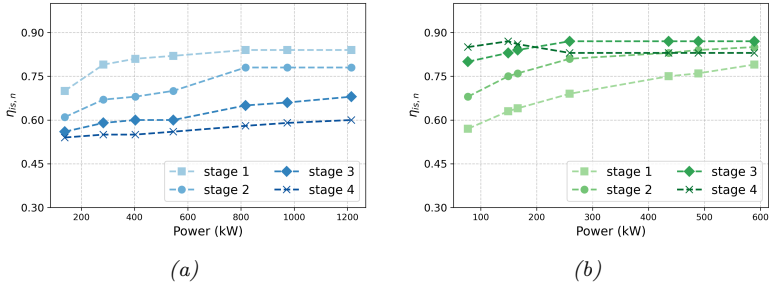


Figure 7.18: Maximum isentropic efficiency of machine stages: (a) compressor, (b) turbine

Table 7.3: Optimized configurations for Zimbabwe for different system sizes

| E_request<br>[MWh/day] | V<br>[m <sup>3</sup> ] | $\dot{m}_{c,n}$<br>[kg/s] | $\dot{m}_{t,n}$<br>[kg/s] | $p_{max}$<br>[bar] | $P_{PV,p}$<br>[kW <sub>p</sub> ] | RTE<br>[%] | LCOE<br>[\$ / kWh] |
|------------------------|------------------------|---------------------------|---------------------------|--------------------|----------------------------------|------------|--------------------|
| 1                      | 67.3                   | 0.19                      | 0.23                      | 92                 | 389                              | 41.5%      | 0.290              |
| 2                      | 124.5                  | 0.40                      | 0.43                      | 101                | 749                              | 46.7%      | 0.256              |
| 3                      | 124.0                  | 0.54                      | 0.47                      | 136                | 1077                             | 47.2%      | 0.236              |
| 4                      | 126.5                  | 0.70                      | 0.72                      | 170                | 1440                             | 47.8%      | 0.228              |
| 6                      | 180.0                  | 1.04                      | 0.95                      | 173                | 2061                             | 50.5%      | 0.214              |
| 8                      | 235.1                  | 1.39                      | 1.20                      | 176                | 2683                             | 52.1%      | 0.207              |
| 10                     | 287.0                  | 1.69                      | 1.38                      | 175                | 3251                             | 53.2%      | 0.202              |

increases, the optimal pressure level progressively rises, reaching a plateau at approximately 175 bar. This behavior can be explained by considering the constraints imposed at small scales. Since the rotational speed cannot be increased beyond its upper limit, maintaining acceptable values of the specific speed  $\omega_s$ , and thus of the efficiency, requires a reduction in the isentropic work. This is achieved by lowering the maximum pressure, which in turn results in smaller stage pressure ratios, less heavily loaded stages, and values of  $\omega_s$  closer to the optimal range. As the machine size increases, this limitation becomes progressively less relevant. From this point onward, the pressure level mainly affects the system from an economic perspective. In particular, the component most sensitive to

pressure variations is the compressed air storage. For the selected vessel volume, the adopted cost function exhibits a minimum in the pressure range between approximately 160 and 180 bar, within which the storage cost is minimized and shows limited sensitivity to pressure variations. Therefore, when pressure no longer has a significant impact on machine performance, there is no need to treat it as an optimization variable. Instead, it can be selected solely on the basis of storage cost minimization, without incurring appreciable penalties in turbomachinery efficiency or overall system performance.

It is worth noting that, in the previous chapter, the optimal pressure was found to be around 117 bar. Under those conditions, the limiting factor was not the loading of the stages, but rather the effect of increasing discharge pressure on reciprocating compressors. Specifically, higher counterpressure leads to an increase in the dead volume, thereby reducing the amount of air that can be effectively delivered to the vessels during the charging phase. For this reason, in the case of piston compressors it was advantageous to limit the charging pressure to values slightly lower than those minimizing the investment cost associated with the storage system.

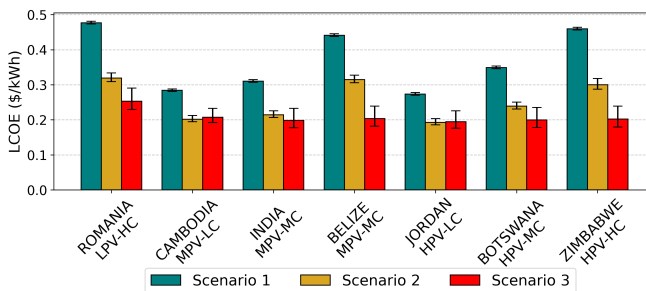


Figure 7.19: LCOE at maximum system size in different geographic sites

Finally, the histogram reported in Figure 7.19 compares the LCOE obtained for the largest system size considered across all seven analyzed locations with the corresponding LCOE values of the diesel-only and

PV+diesel configurations. The results indicate a general reduction in LCOE for the third scenario when turbomachinery is adopted, compared to the case based on volumetric machines, previously shown in Figure 6.20. Consequently, the introduction of the CAES system becomes economically advantageous in a larger number of locations than in the previous case. Only in locations characterized by the lowest diesel fuel costs the CAES-based solution fails to be competitive with respect to a configuration relying solely on photovoltaic generation without energy storage.

## 7.7 Conclusions

This chapter investigated the scalability of hybrid PV-diesel-CAES systems by extending the analysis to bigger system scale, compared to the previous chapter, and by replacing volumetric machines with radial turbomachinery in the compression and expansion stages. The primary objective was to assess whether this technological transition can improve both the energetic and economic performance of CAES-based off-grid systems under realistic operating conditions.

To this end, a performance model for turbomachinery based on non-dimensional characteristic maps was developed, including a realistic scaling of maximum achievable isentropic efficiency and a technological constraint on maximum rotational speed. Particular attention was devoted to control strategies capable of following the available charging power and the load demand, through the use of inlet guide vanes and variable stator vanes. Optimal scheduling laws were derived through a dedicated off-design optimization procedure for the compressor and turbine regulation systems, reformulated into non-dimensional look-up maps, and embedded in the full plant model to enable efficient and consistent system-level simulations over a wide operating range.

The proposed framework was applied to a case study representative of favorable off-grid conditions, with Zimbabwe selected as reference location, and an economic optimization was performed by minimizing the levelized cost of electricity (LCOE). The results allow for a direct

comparison between turbomachinery-based and volumetric-based CAES configurations and provide quantitative insights into the impact of system scale on performance and cost.

The main outcomes of the chapter can be summarized as follows:

- A turbomachinery performance model was developed based on consolidated map-based formulations from the literature, enriched with realistic scale-dependent efficiency ceilings and speed constraints, and systematically coupled with automated off-design control schedules embedded in the plant model. This modeling framework represents a practical and original contribution for the performance assessment of small-scale CAES systems.
- The LCOE exhibits a clear decreasing trend with system scale, following an approximately hyperbolic behavior. For the Zimbabwe case study, it decreases from about 0.29 \$/kWh at small sizes to approximately 0.20 \$/kWh at 10 MWh/day.
- When comparing turbomachinery-based and volumetric-based CAES solutions, the corresponding LCOE curves intersect at around 3 MWh/day. Below this threshold, volumetric machines remain economically preferable, while above it radial turbomachinery becomes the most cost-effective option.
- Unlike systems based on volumetric machines, whose round-trip efficiency remains approximately constant, the turbomachinery-based configuration shows a marked increase in round-trip efficiency with scale, rising from approximately 41% for small systems up to about 54% for the largest configurations considered. The improvement in round-trip efficiency is primarily attributed to the increase in isentropic efficiency achievable by larger turbomachines. At small scales, performance is penalized by the imposed maximum rotational speed constraint, which prevents operation at optimal specific speed conditions.

- The optimal maximum storage pressure increases with system size and converges toward values around 175 bar. For smaller systems, lower pressures are preferable in order to limit efficiency losses associated with off-design operation of the turbomachinery. The convergence of the optimal pressure is explained by the minimum of the pressure vessel cost curve.
- The adoption of turbomachinery systematically reduces the LCOE across all analyzed locations, expanding the range of scenarios in which CAES represents a competitive solution for off-grid hybrid systems, with the exception of cases characterized by very low diesel fuel costs.

Overall, this chapter demonstrates that the transition from volumetric machines to radial turbomachinery constitutes an effective pathway to enhance the performance of CAES systems, particularly at medium-to-large storage scales. While volumetric solutions remain preferable at small sizes, turbomachinery enables substantial gains in round-trip efficiency and economic competitiveness beyond a scale of a few hundred kW. At the same time, the results underline the importance of explicitly accounting for the degradation of turbomachinery efficiency as plant size decreases, as neglecting this effect may lead to overly optimistic performance estimates at small scales. Moreover, the identified trends in optimal pressure levels provide practical design guidelines for further analysis.



## Chapter 8

# Techno-Economic Assessment of Grid-Connected Small-Scale CAES Systems

This chapter aims to provide an additional perspective on the potential deployment of small-scale CAES systems by extending the analysis to slightly larger capacities, in the range of 1–10 MW, and by shifting the focus from off-grid applications to grid-connected configurations. In this context, the chapter offers a preliminary investigation into how systems of this type may be integrated into a networked environment and whether such integration could prove economically advantageous.

To this end, the model developed in the previous chapter is revisited and embedded within a grid-connected framework, where the system is assumed to operate through price-arbitrage mechanisms. As this aspect does not represent the primary focus of the dissertation, the analysis adopts a simplified approach aimed at providing high-level insights and broadening the overall assessment, rather than delivering a detailed treatment of market regulation. A comprehensive discussion of market rules and regulatory frameworks lies beyond the scope of this work and may represent a potential avenue for future research.

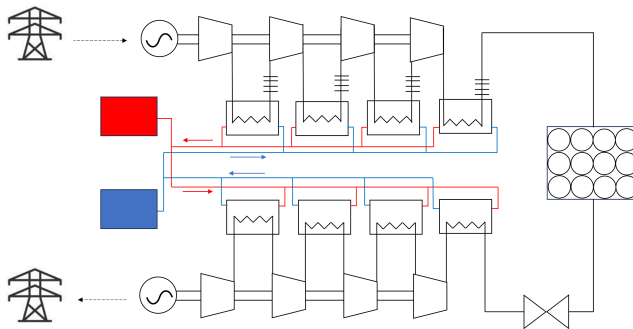


## 8.1 System description

The system considered in this chapter retains the same overall configuration as that analyzed in the previous chapter, based on the use of radial turbomachinery. The main difference lies in the fact that, in this case, the system is not connected to a dedicated generation source or a specific end user, but is instead directly connected to the electrical grid. This configuration allows the system to exchange energy with the grid during both the charging and discharging phases through the purchase and sale of electricity.

Furthermore, as will be discussed in greater detail in the following sections, the storage system considered in this analysis may be implemented using different technological solutions. In particular, compressed air storage is realized either through pressure vessels or through pipelines, depending on the specific case examined. This choice influences the economic evaluation and the constraints associated with the maximum allowable pressure; however, it does not entail any changes to the underlying thermodynamic model of the CAES system.

Figure 8.1 provides a schematic representation of the system considered.



*Figure 8.1: Plant schematic in grid-connected configuration*

## 8.2 Methodology

The storage system is assumed to operate primarily under an electricity price arbitrage strategy, i.e. by purchasing electricity from the grid during periods of low market prices and selling it back when prices are higher. While this operating mode provides a direct economic benefit to the storage operator, it also performs a system-level function by shifting electrical load in time.

From the perspective of the power system, price-arbitrage driven operation results in increased electricity demand during off-peak periods and a reduction of net demand during peak periods, thereby contributing to load shifting. This behaviour can support grid operation by mitigating peak loads, improving asset utilisation, and facilitating the integration of variable renewable energy sources.

Based on these considerations, the control strategy adopted in this work aims at reproducing the characteristic operational behaviour of a price-arbitrage-driven storage system, while maintaining a modelling framework that is computationally efficient and compatible with detailed thermo-mechanical system simulations.

### 8.2.1 Simplified charge–discharge logic for price-arbitrage operation

An energy storage system performing electricity price arbitrage would ideally be modelled through a long-term simulation explicitly accounting for time-varying electricity prices and price-based decision thresholds. Such an approach, however, would significantly increase the computational burden and is not strictly required to capture the dominant thermodynamic and economic effects relevant to the present study.

For this reason, a simplified control strategy is adopted, which nonetheless allows the behaviour of a price-arbitrage-driven storage system to be realistically mimicked. Rather than explicitly modelling market prices, predefined operational windows are introduced through the use of boolean control variables. These variables enable or disable the charging and

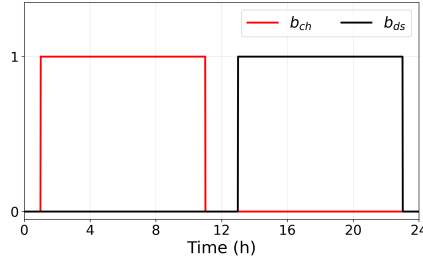


Figure 8.2: Boolean activation signals for charging and discharging phases

discharging phases within given time intervals, such variables are graphically illustrated in Figure 8.2. The use of two separate representative time windows for charging and discharging is a reasonable assumption, as electricity markets typically exhibit clustered and well-separated periods of low and high prices, rather than a rapid alternation between favourable and unfavourable conditions, as will be further illustrated in the following section.

The temporal windows do not correspond to actual market hours and charging and discharging are therefore decoupled from instantaneous electricity prices. As shown in Figure 8.2, the maximum duration of each charging and discharging window is set to 10 hours. These values represent upper bounds rather than imposed operating times: the system is not forced to operate for the entire duration of the window. The economic assessment is instead performed by assigning an average electricity cost to the energy exchanged with the grid, as further discussed in the following sections.

The two boolean variables are defined as follows:

$$b_{\text{ch}}(t) \in \{0, 1\}, \quad b_{\text{dis}}(t) \in \{0, 1\}, \quad (8.1)$$

which indicate whether the charging or discharging phase is allowed at time  $t$ .

The operation of the storage system is constrained by the pressure inside the storage tank. The charging phase is activated if and only if the

corresponding boolean variable is equal to one and the pressure inside the storage tank is lower than the maximum admissible value:

$$\text{Charging active if } b_{\text{ch}}(t) = 1 \wedge p(t) < p_{\text{max}}. \quad (8.2)$$

Analogously, the discharging phase is activated if and only if the corresponding boolean variable is equal to one and the pressure inside the storage tank is higher than the minimum admissible value:

$$\text{Discharging active if } b_{\text{dis}}(t) = 1 \wedge p(t) > p_{\text{min}}. \quad (8.3)$$

Accordingly, the charging phase is terminated either when the maximum storage pressure is reached or when the maximum charging window duration of 10 hours is exceeded. Similarly, the discharging phase ends either when the minimum storage pressure is reached or after a maximum duration of 10 hours. The numerical values of  $p_{\text{max}}$  and  $p_{\text{min}}$  adopted in the present analysis are specified and discussed in next sections.

During the charging phase, the compression system is activated. The electrical power absorbed from the grid is not imposed directly. Instead, the inlet guide vane angle  $\alpha$  of the first compression stage is imposed and selected so as to maximise the overall compression efficiency, based on the current backpressure. This approach minimizes the electrical energy drawn from the grid, and therefore the associated cost, for a given amount of stored air. An alternative control strategy could consist in maximizing the compression power in order to concentrate the charging process within the lowest-cost hours; however, preliminary analyses showed that operation close to the maximum efficiency provides better economic performance, and this approach was therefore adopted in the present work. This strategy is intended to define a priori the regulation of the first IGV angle, while the regulation of the subsequent stages is determined accordingly, as discussed in the previous chapter. As will be detailed in the following sections, the overall sizing of the system is instead carried out by maximising its economic return. Defining the IGV regulation in advance therefore allows the optimisation procedure to be simplified, by excluding this aspect from

the economic optimisation.

The electrical power required to satisfy this operating condition therefore results as an output of the model:

$$P_c(t) = P_c(\alpha_{ch}^*(t), \mathbf{x}(t)). \quad (8.4)$$

where  $\alpha^*$  represents the optimum IGV angle and  $\mathbf{x}(t)$  denotes the vector of system state variables.

The optimisation of the inlet guide vane angle follows a procedure analogous to that described in Section 7.4. Rather than allowing all vane angles to vary independently, only the inlet guide vane angle of the first compression stage is treated as a free optimisation variable. The remaining vane angles are instead determined through the optimal relationships previously derived.

An analogous control logic is applied during the discharging phase. When the discharging phase is active, the inlet guide vane angle of the expander is selected to maximise the expansion efficiency and the electrical power delivered to the grid is obtained as:

$$P_{exp}(t) = P_{exp}(\alpha_{dis}^*(t), \mathbf{x}(t)). \quad (8.5)$$

Adopting the convention that positive power corresponds to electricity absorbed from the grid, the net power exchanged with the grid can be written in a compact form as:

$$P_{grid}(t) = \begin{cases} P_{comp}(t), & \text{if } b_{ch}(t) = 1 \wedge p(t) < p_{max}, \\ -P_{exp}(t), & \text{if } b_{dis}(t) = 1 \wedge p(t) > p_{min}, \\ 0, & \text{otherwise.} \end{cases} \quad (8.6)$$

Although the proposed control strategy does not explicitly rely on instantaneous electricity prices, it reproduces the characteristic system-level behaviour of price arbitrage by shifting energy uptake and release across predefined operational windows. This approach significantly reduces modelling complexity while preserving the essential coupling between

storage operation, component efficiency, and grid interaction.

## 8.2.2 Economic analysis

### 8.2.2.1 Performance Indices

The main economic indicator considered in this analysis is the Net Present Value (NPV), defined as the sum of discounted cash flows over the entire lifetime of the investment. The NPV is expressed as

$$\text{NPV} = \sum_{i=0}^Y \frac{CF_i}{(1+d)^i} \quad (8.7)$$

where  $CF_i$  denotes the net cash flow in year  $i$ ,  $d$  is the discount rate, and  $Y$  represents the total number of years considered in the system lifetime.

The structure of the cash flows reflects the economic transactions associated with the operation of a grid-connected energy storage system. In the initial year ( $i = 0$ ), the cash flow is equal to the total capital investment cost:

$$CF_0 = -CAPEX \quad (8.8)$$

For each subsequent year ( $i > 0$ ), the annual cash flow is defined as

$$CF_i = R_i^{\text{sell}} - C_{\text{ch},i} - O\&M \quad (8.9)$$

where  $R_i^{\text{sell}}$  represents the revenues obtained from selling electricity to the grid,  $C_{\text{ch}}$  is the cost associated with the charging phase of the storage system, and  $O\&M$  denotes the annual operation and maintenance costs.

The charging cost is computed as

$$C_{\text{ch}} = c_{el,ch} \cdot E_{in} \quad (8.10)$$

where  $E_{in}$  is the annual electrical energy used to charge the storage system and  $c_{el,ch}$  represents the average cost of the electricity withdrawn from the grid for charging purposes.

Similarly, the revenues from electricity sales are expressed as

$$R_i^{\text{sell}} = c_{el,\text{sell}} \cdot E_{out} \quad (8.11)$$

where  $E_{out}$  denotes the useful electrical energy delivered by the storage system to the grid, and  $c_{el,\text{sell}}$  is the average electricity price associated with energy injected into the grid. Both  $c_{el,\text{ch}}$  and  $c_{el,\text{sell}}$  are assumed to be average values over the considered period; a detailed description of their derivation and the adopted pricing assumptions is provided in the following sections.

This formulation enables a comprehensive evaluation of the economic performance of grid-connected storage systems by explicitly accounting for market-based energy arbitrage and operational expenses over the entire lifecycle. A positive NPV indicates that the investment is economically viable under the assumed conditions. Furthermore, the cumulative discounted cash flow allows for the identification of the break-even point (BEP), defined as the year in which the NPV becomes non-negative for the first time, corresponding to the recovery of the initial investment. To ensure consistency and comparability with previous analyses, the discount rate, maintenance costs, and system lifetime are assumed to be identical to those adopted in the preceding sections.

#### 8.2.2.2 Electricity cost

Two potential installation locations are considered in this work, namely Chile and Denmark, selected on the basis of the temporal variability of electricity prices observed in their respective power systems. As will be discussed in detail in the following, both countries exhibit pronounced intra-day price fluctuations, making them favourable case studies for the deployment of energy storage systems operating under a price-arbitrage strategy. Chile, in particular, represents an extreme case, characterised by electricity price patterns that are rarely observed elsewhere, whereas Denmark, while still exhibiting significant price variability, constitutes a more representative and realistic reference case in a broader international

context.

For both locations, since the analysis is aimed at identifying typical techno-economic trends rather than capturing rare price spikes, a representative-year profile is adopted. The representative year was constructed by averaging the hourly electricity prices of 2023 and 2024, using data derived from [275]. The resulting hourly electricity price profiles for each day of the representative year are shown in Figure 8.3. This approach allows reducing the influence of year-specific anomalies, while retaining the characteristic daily and seasonal price patterns relevant for techno-economic assessments. In addition, Figure 8.4 provides a detailed view of the first week of the representative year, to underline the daily and intra-daily price patterns.

In the Chilean case, a marked contrast can be observed between the central hours of the day, during which electricity prices frequently approach zero, and the early morning and evening hours, when prices can frequently exceed 150 \$/MWh. This behavior represents an extreme manifestation of a trend that is common to many power systems with a high penetration of variable renewable energy sources, and is primarily driven by the large contribution of solar generation, which leads to substantial oversupply during daylight hours. A pronounced seasonal effect is also evident for Chile, with the first months of the year characterized by generally lower electricity prices, including during night-time hours. This behavior can be attributed to the austral summer period, during which solar availability is enhanced and overall electricity demand patterns differ from those observed in winter months.

Denmark, by contrast, exhibits a comparatively smoother price profile, although it remains one of the countries worldwide with the highest degree of electricity price volatility, including periods in which prices approach zero. This behaviour is largely associated with the high penetration of renewable energy sources in the Danish power system, in particular wind power, complemented by solar generation. From the weekly profiles shown in Figure 8.4, it can be observed that periods of high and low cost are effectively clustered into concentrated time windows and exhibit a 24-hour



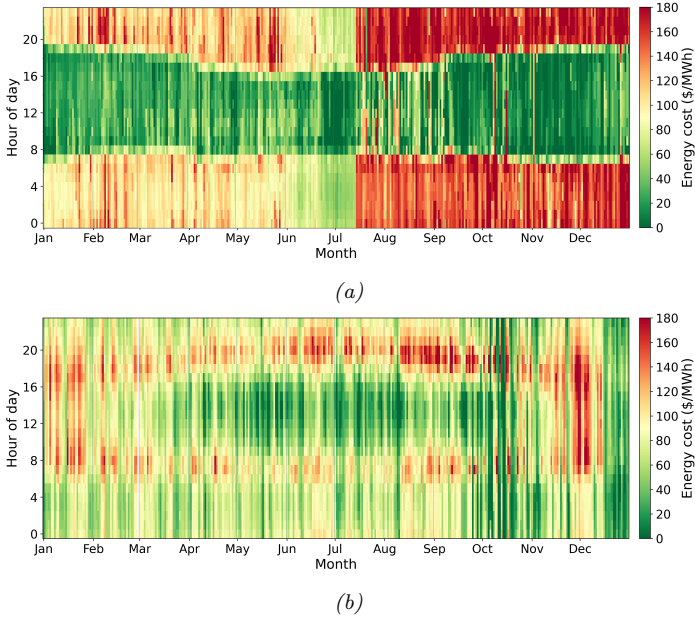


Figure 8.3: Hourly electricity prices for each day of the representative year for: (a) Chile and (b) Denmark. Data derived from [275].

repeatability. This behavior justifies the adopted approach, which is based on the definition of one charging window and one discharging window over a 24-hour horizon.

The electricity price data for both locations were analyzed with the aim of extracting representative trends suitable for simplified modeling analyses. The hourly price time series over the entire year was first organized on a daily basis. Then, for each day, the analysis focused on identifying the most expensive and least expensive operating periods. Specifically, for a given number  $n$  of hours, the  $n$  highest and the  $n$  lowest hourly prices of the day were selected. These daily values were then averaged over the whole year, yielding annual reference costs associated with operation during expensive and cheap hours, respectively. The trends

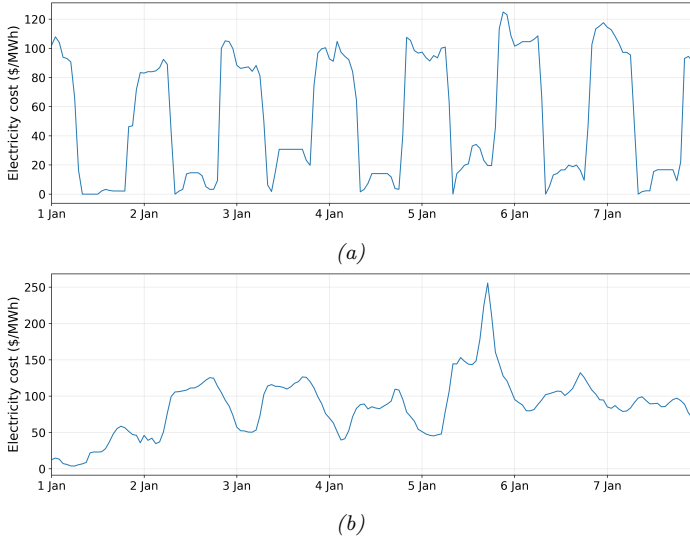
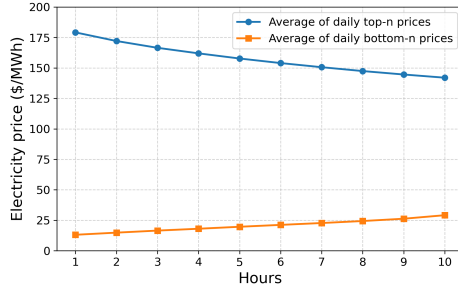


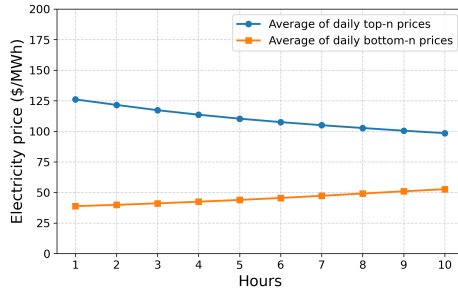
Figure 8.4: Hourly electricity prices for the first week of the representative year for: (a) Chile and (b) Denmark. Data derived from [275].

obtained by parameterizing the value of  $n$ , reported in Figure 8.5, provide a compact and intuitive representation of typical high and low-price electricity costs as a function of the considered time window, suitable for use in the simplified modelling approaches described.

As described in the previous section, the storage system operation is not scheduled on the basis of instantaneous electricity prices. Instead, the system is simulated assuming simplified and repetitive charging and discharging cycles. For each simulated configuration, the average electricity purchase and selling prices are assigned a posteriori based on the effective duration of the charging and discharging phases resulting from the dynamic simulation. In particular, the charging and discharging durations,  $t_{ch}$  and  $t_{dis}$ , are extracted from the representative operating cycle of each configuration. These durations correspond to the effective number of hours during which the storage system absorbs or delivers electricity to the grid, respectively. The average electricity cost for charging,  $c_{el,ch}$ , is



(a)



(b)

Figure 8.5: Average electricity prices of the  $n$  highest and lowest-cost operating hours for: (a) Chile and (b) Denmark.

then defined as the mean value of the  $t_{\text{ch}}$  lowest hourly electricity prices of the representative year. Analogously, the average selling price,  $c_{\text{el,sell}}$ , is defined as the mean value of the  $t_{\text{dis}}$  highest hourly electricity prices of the same year. Operationally, this corresponds to selecting, from the curves shown in Figure 8.5, the average electricity price associated with  $t_{\text{ch}}$  hours for charging and with  $t_{\text{dis}}$  hours for discharging. The resulting prices are configuration-dependent and directly reflect the different operating times associated with each optimal design. This procedure ensures a consistent coupling between the simulated operating behaviour of the storage system and the economic evaluation, while avoiding the explicit modelling of time-varying electricity prices within the dynamic simulation.

### 8.2.2.3 Economic feasibility enhancement scenarios

From an economic perspective, this analysis investigates two complementary aspects that may enhance the overall economic feasibility of the proposed CAES system. Starting from a baseline configuration, these aspects are introduced progressively in order to quantify their impact on the economic performance indicators. The first aspect focuses on cost reduction strategies related to the air storage subsystem, while the second examines the potential role of regulatory support mechanisms in improving project profitability. A detailed description and justification of the analyzed scenarios is provided in the following.

- **Reuse of natural gas pipelines as air storage**

As extensively discussed in the previous chapters, the air storage system represents a significant share of the overall plant cost; therefore, it is of particular interest to investigate alternative configurations that may reduce this contribution. To this end, a scenario is considered in which the CAES system is coupled with the reuse of existing natural gas pipelines as an air storage medium. In this framework, the reuse of decommissioned gas pipelines is investigated as a potential solution to lower the cost associated with compressed air storage.

In the context of an ongoing energy transition and increasing decarbonization, it is reasonable to expect that a substantial fraction of the current natural gas transmission infrastructure will be progressively decommissioned, thus representing a potentially valuable asset for repurposing [276]. For this reason, the reuse of pipeline infrastructure has attracted considerable attention in the literature, particularly in hydrogen-dominated decarbonized energy systems, where several studies have assessed the technical feasibility, economic performance, and cost implications of repurposing natural gas pipelines for hydrogen transport [276, 277], as well as for hydrogen storage applications [278]. These works highlight, on the one hand, the technical challenges associated with hydrogen transport,

such as material embrittlement and leakage issues, and, on the other hand, the potential economic benefits arising from the exploitation of existing infrastructure. Although more limited in number, similar concepts have also been proposed for CAES systems. In particular, some studies suggest the use of existing natural gas pipelines as compressed air energy storage reservoirs in order to overcome the high costs associated with aboveground storage solutions [279]. Other innovative approaches investigate the reuse of abandoned oil wells as underground air storage tanks for CAES applications [280]. Based on these considerations, it is therefore of interest to assess how the overall economic performance of the CAES system changes when existing infrastructures are reused.

In particular, large-diameter pipelines with a nominal diameter of 36" and a maximum allowable operating pressure of 80 bar are considered, which represent realistic values for high capacity natural gas transmission pipelines [281].

In the scenario analyzed in this work, the capital cost associated with pipeline acquisition is neglected, assuming that the infrastructure is already available, and only the costs related to maintenance and adaptation required for compressed air storage operation are taken into account. Based on the analysis presented in [277], an investment cost of approximately 2.3 M\$/km is assumed for a standard pipeline. Operation and maintenance costs are typically estimated as about 1% of the initial investment cost. Accordingly, since only the maintenance phase is considered in this study, an effective maintenance cost of approximately 23\$/m per year is assumed.

- **Incorporation of regulatory incentives into the economic framework**

Another relevant aspect that deserves consideration concerns the possibility of regulatory intervention through incentive-based mechanisms. As highlighted by Miller and Carriveau [282], the current deployment of energy storage systems is constrained more by project

financing issues than by technological limitations. High upfront capital costs, together with a lack of dedicated financing instruments and stable incentive schemes, significantly hinder the realization of large-scale storage projects. This challenge is particularly pronounced for capital-intensive technologies such as CAES. Matos et al. [283] emphasize that the removal of existing regulatory barriers, along with the introduction of broader and more coherent policy frameworks, constitutes a key enabling factor for the development of energy storage technologies. Empirical evidence further suggests that the diffusion of CAES projects is more pronounced in countries where energy storage is already well integrated into electricity markets and supported by targeted policy measures, such as the United States.

More generally, several incentive schemes for energy storage have been introduced worldwide in recent years [282, 284], primarily driven by the potential of ESS to enhance power system flexibility and reduce greenhouse gas emissions. In the U.S. context, the Energy Storage System tax incentive bill, proposed in 2016, aimed at granting tax credits to storage systems with a minimum capacity of 5 kWh, independently of the underlying technology. The incentive scheme encompasses a wide range of storage typologies (chemical, electrochemical, mechanical, thermal, and electrical), provided that the system is capable of charging energy and discharging it when required. In California, additional incentive programs are in place that allow end users to receive a rebate of up to 30% of the installed storage system cost, with a particular focus on behind-the-meter batteries for self-consumption applications [285].

Incentive mechanisms are also evolving over time. The United Kingdom represents a notable example, having implemented concrete and operational support schemes for energy storage through market-based instruments. Since 2014, storage assets have been allowed to participate in the Capacity Market, where capacity is procured via competitive auctions and remunerated on a £/MW/year basis

[286]. More recently, a cap-and-floor regulatory regime has been introduced for long-duration energy storage (LDES), aimed at guaranteeing a minimum revenue stream while limiting excessive returns [287]. This framework significantly reduces revenue uncertainty and investment risk, thereby facilitating the deployment of storage technologies characterized by long discharge durations.

In Italy, a similar approach has only been adopted very recently with the introduction of a dedicated storage capacity mechanism. The Storage Forward Capacity Market (MACSE), defined by Terna, is a market-based auction scheme designed to procure electricity storage capacity through long-term contracts, with the objective of increasing system flexibility and supporting renewable energy integration [288]. Based on the results of the first auctions, storage developers are expected to receive an average remuneration of nearly 15000\$/MW per year for the capacity made available to Terna, which can be approximately translated into 50–70 \$/MWh of discharged energy depending on the actual number of annual charge–discharge cycles performed [289].

Against this regulatory background, this study investigates the impact of a simplified incentive scheme based on a feed-in premium (FIP) approach, whereby an additional remuneration is granted for each unit of electrical energy injected into the grid by the storage system. To ensure realism, mimic typical real-world incentive designs, and avoid unrealistic optimization outcomes, the eligibility for the premium is limited in time. In particular, the FIP is applied exclusively to the first four hours of discharge, chosen to represent typical evening peak-demand conditions, as it would be unrealistic to assume the presence of incentive mechanisms outside periods of system stress or peak demand. Outside this time window, the storage system is still allowed to sell energy on the market, but without receiving any additional remuneration.

From a modeling perspective, the annual revenues from electricity

sales in year  $i$  are expressed as:

$$R_i^{\text{sell}} = c_{el, \text{sell}} \cdot E_{out} + \text{FIP} \cdot E_{out}^{\text{inc}}, \quad (8.12)$$

where  $c_{el, \text{sell}}$  is the average electricity price associated with energy injected into the grid,  $E_{out}$  denotes the total annual electrical energy delivered by the storage system, FIP represents the feed-in premium expressed in \$/MWh, and  $E_{out}^{\text{inc}}$  is the fraction of discharged energy eligible for incentivization. This approach allows the economic effect of peak-oriented regulatory support to be consistently evaluated within the lifecycle-based investment analysis, without altering the assumptions on charging costs, operation and maintenance expenses, discount rate, or system lifetime adopted in the baseline scenario.

### 8.2.3 Analyses performed and optimization procedure

The system described in the previous sections has been analysed by integrating it within an optimization framework aimed at determining its optimal sizing. As discussed earlier, no predefined charging or discharging duration was imposed. Moreover, the cost associated with system operation varies as a function of the charging and discharging time. For this reason, the introduction of an optimization procedure is essential in order to identify the optimal system size, which in turn determines an optimal storage duration driven by the temporal evolution of electricity prices.

Consistently with the approach adopted in the previous chapters, all optimizations are based on the EGO algorithm. In this study, the objective function is the NPV evaluated at the end of the system lifetime. Consequently, the optimization problem is formulated as a maximization problem, with the goal of identifying the system configuration that maximizes the economic value of the investment over its lifecycle.

In particular, the maximum operating pressure is prescribed a priori. For configurations employing pressure vessels as the storage system, the pressure is fixed at 175 bar, in accordance with the lessons learned from the previous chapter. Conversely, for configurations based on pipelines,



the maximum pressure is limited to 80 bar. The minimum pressure is fixed at  $p_{\min} = 5$  bar in order to ensure a sufficient minimum inlet pressure at the expander. From a purely performance-oriented perspective, operating over a narrower pressure range would be beneficial, limiting the off-design of the turbomachinery. However, a wider pressure range is deliberately adopted in this work to minimise the required storage volume. As shown in the previous chapters, the cost associated with the storage system represents a significant fraction of the total system cost, and reducing the storage volume leads to a more favourable economic outcome.

At large system scales, the use of pressure vessels implies very large overall storage volumes. Under these conditions, adopting a large number of small, standardized pressure bottles may no longer represent the most convenient solution from a plant design perspective. For high total storage capacities, it may instead be advantageous to consider fewer vessels with significantly larger individual volumes, as this can reduce system complexity, limit the extent of interconnection piping, valves, and manifolds, and simplify both installation and maintenance. From an economic perspective, the estimation of storage costs in this regime becomes inherently uncertain. In fact, the total cost is influenced not only by the pressure-containing elements, but also by the balance-of-plant components. Available cost correlations in the literature for pressurized air storage show a wide dispersion, with reported values ranging from approximately 1000 to more than 8000 \$/m<sup>3</sup> [49, 290]. In the present work, consistency with the cost modelling adopted in the previous chapters is preserved. Assuming a maximum pressure of 175 bar and a standard vessel diameter of 0.229 m, the cost correlation previously employed yields an estimated storage cost of approximately 4100 \$/m<sup>3</sup>. This value is therefore adopted throughout the analysis as a representative average storage cost. With this assumption, the analysis deliberately abstracts from the specific choice between multiple small pressure bottles and fewer large-volume storage vessels, effectively treating the storage system as a pressurized volume characterized by an average cost per unit volume.

Furthermore, the nominal power of the turbine is prescribed and

parametrized at three discrete levels (i.e. 1 MW, 5 MW, and 10 MW).

Under these assumptions, the optimization variables are limited to the storage volume and the compressor power. The analyses are conducted for the two different geographical locations and for each of the three power levels considered. For each case, both storage configurations (i.e. vessel-based and pipeline-based) are analysed. In the case of Denmark, an additional scenario including economic incentives is also investigated, as will be discussed in detail in the results section.

The ranges of variation of the optimization variables are reported in Table 8.1 for the three power levels and for the different storage configurations considered.

For each optimization case, the thermodynamic model of the CAES system is simulated by repeating consecutive charging and discharging cycles, with the initial conditions of each cycle set equal to the final state of the previous one. The simulation is continued until a cyclic steady-state condition is reached, defined by equality between the initial and final storage pressure of the cycle. The performance indicators of the final simulated cycle are then averaged and assumed to be representative of the entire year.

*Table 8.1: Ranges of the optimization variables for the different storage configurations and turbine power levels*

| Variable                         | Symbol | Range of variation   |         |         |                        |          |           |
|----------------------------------|--------|----------------------|---------|---------|------------------------|----------|-----------|
|                                  |        | Vessel-based storage |         |         | Pipeline-based storage |          |           |
|                                  |        | 1 MW                 | 5 MW    | 10 MW   | 1 MW                   | 5 MW     | 10 MW     |
| Compressor rated power [MW]      | $P_c$  | 0.1–10               | 0.5–50  | 1–100   | 0.1–10                 | 0.5–50   | 1–100     |
| Storage volume [m <sup>3</sup> ] | $V$    | 8–500                | 40–2500 | 80–5000 | 20–1500                | 100–7500 | 200–15000 |

## 8.2.4 Summary of the methodological approach

Before turning to the results, it is useful to summarize the different methodological steps described in the previous sections and to clarify the rationale behind the main modelling choices.

The overall workflow can be summarized as follows:

- **Thermodynamic model with boolean activation windows** (Section 8.2.1). The system is modeled through a thermodynamic model where charging and discharging phases are enabled by two boolean variables associated with predefined daily time windows that define the maximum allowable charging/discharging time. The thermodynamic model is run through consecutive charge–discharge cycles until cyclic steady state is reached.
- **Pre-processing of electricity prices into cost–duration curves** (Section 8.2.2.2). For the analysed site, the hourly electricity prices of a representative year are reorganized into curves expressing the average cost of the  $n$  cheapest and  $n$  most expensive hours of the day, averaged over the year.
- **NPV-based economic evaluation** (Sections 8.2.2.1). The economic performance of the system is assessed by computing the Net Present Value. The charging and selling prices,  $\bar{c}_{el,ch}$  and  $\bar{c}_{el,sell}$ , considered during the economic analysis are assigned a posteriori, on the basis of the operating durations  $t_{ch}$  and  $t_{ds}$  resulting from the dynamic simulation and by querying the curve described in the previous point.
- **NPV-maximizing sizing optimization through EGO** (Section 8.2.3). The storage volume  $V$  and the rated compressor power  $P_c$  are treated as the decision variables of the sizing optimization problem. For each candidate configuration, the thermodynamic model provides the system energy performance and the corresponding operating durations  $t_{ch}$  and  $t_{ds}$ . These are used to interrogate the cost–duration curves and determine the associated average charging and selling electricity prices, which are then employed in the economic analysis. The resulting end-of-life Net Present Value is finally used as the objective function of the EGO optimization algorithm.

Figure 8.6 provides a graphical synthesis of the overall workflow. The diagram distinguishes the pre-processing step that generates the cost–duration curves (left) and the iterative sizing-optimization loop (right). The two procedures are linked by the operating durations  $t_{ch}$  and  $t_{ds}$  extracted from the dynamic simulation, which are used to interrogate the cost–duration curves and retrieve the corresponding average electricity prices, in turn fed back to the economic analysis.

The rationale behind the proposed methodology is to reproduce, in a relatively simple and computationally efficient way, the characteristic behaviour of a price-arbitrage-driven storage system while keeping the dynamic simulations tractable within an optimization framework. A fully detailed economic assessment would require simulating the storage operation over an entire representative year while explicitly accounting for time-varying electricity prices. In such a formulation, the charging and discharging phases would typically be activated according to a price threshold, which would itself become an additional optimization variable. Moreover, coupling a detailed dynamic model with a fully time-resolved annual simulation can become computationally expensive. For this reason a common strategy consists in compressing the annual profiles into a limited number of representative days through statistical techniques such as clustering, thereby reducing the simulation time while preserving the main statistical features of the price signal. The approach proposed in this work follows a different and simpler route, aimed not only at reducing the computational cost of the simulations but also at decreasing the number of optimization variables. The average charging and selling electricity prices are assigned a posteriori based on the operating durations obtained from the dynamic simulations. In practice, this is equivalent to assuming that, for each day of the representative year, the system charges during the  $t_{ch}$  cheapest hours and discharges during the  $t_{ds}$  most expensive ones, with these durations inherited directly from the cyclic steady-state simulations. In this way, the price threshold governing the system operation is not introduced explicitly, but is implicitly determined by the procedure and naturally adapts to the system size. As a consequence, the charging

and discharging durations are not treated as independent optimization variables. They emerge instead from the thermodynamic behaviour of the system, which depends on the storage volume and on the rated power of the compression and expansion trains, i.e. the actual decision variables of the sizing problem. The resulting operating durations are then used to interrogate the cost-duration curves and determine the corresponding average electricity prices for the economic assessment. Finally, the end-of-life Net Present Value is used as the objective function of the EGO optimization algorithm. This formulation removes one degree of freedom that would otherwise need to be optimized explicitly, significantly reducing the computational cost while still capturing the main techno-economic trade-offs relevant to the scope of the present work. A more refined formulation, based on explicit threshold-based control coupled with fully time-resolved annual simulations, is left for future investigations.

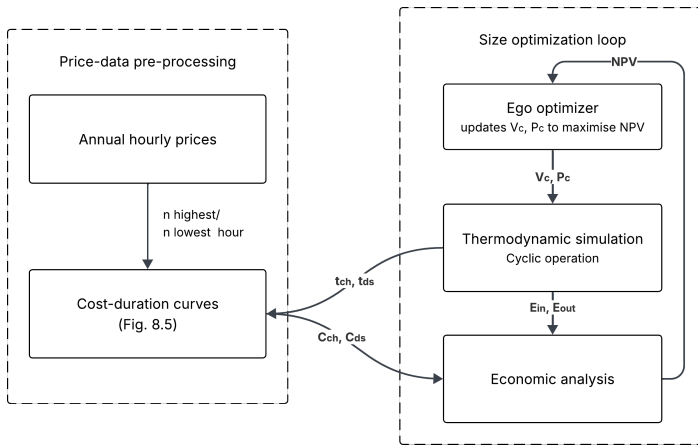


Figure 8.6: *Schematic representation of the overall methodological workflow*

### 8.3 Results

Starting from the configuration based on vessel-based air storage, the results are presented in Figure 8.7, which reports the cumulative NPV of the optimized configurations for the Chilean case as a function of the selected output power levels. The corresponding design parameters and key performance indicators are summarized in Table 8.2. An analysis of Figure 8.7 shows that only the system with an output power of 10 MW achieves a positive economic return; however, the associated break-even point occurs at approximately 22 years, indicating a relatively unattractive return on investment. The remaining configurations exhibit cumulative NPV values that remain consistently below zero over the entire project lifetime.

The data reported in the table indicate an increase in round-trip efficiency compared to the configurations analyzed in the previous chapter, mainly attributable to the further upscaling of the machinery, which results in higher component efficiencies. In addition, in this configuration both the charging and discharging phases operate without being constrained by available or required power profiles. This operating strategy significantly reduces off-design conditions for the components, thereby improving the overall system efficiency.

When considering the Danish electricity price profile, the system fails to generate a positive cash flow, as the limited price spread between off-peak and peak periods provides insufficient revenue to offset energy conversion losses and non-negligible operation and maintenance costs. Consequently, the optimization process tends to minimize the system size in order to reduce CAPEX, and the resulting configurations and NPV values are not reported, as they are not meaningful.

In conclusion, this storage solution appears unlikely to be economically advantageous under price-arbitrage operation, becoming potentially feasible only in contexts characterized by electricity prices with a pronounced spread between off-peak and peak periods. This outcome highlights the need to investigate improved or alternative solutions.

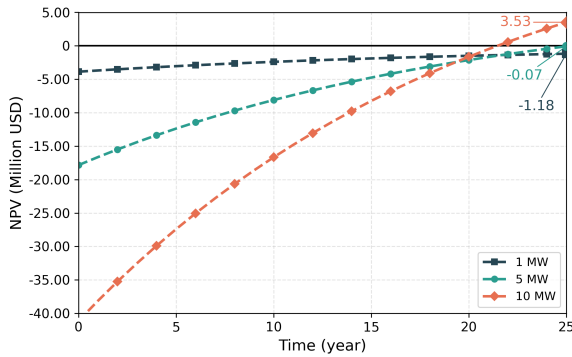


Figure 8.7: Cumulative NPV for the Chilean location with vessel based storage

Table 8.2: Main sizing and performance parameters of the optimized vessel-based storage configurations for the Chilean case study

| $P_t$<br>[MW] | $P_c$<br>[MW] | $V$<br>[m <sup>3</sup> ] | RTE<br>[%] | $t_{ch}$<br>[h] | $t_{ds}$<br>[h] | $E_{out}$<br>[ $\frac{MWh}{day}$ ] | NPV<br>[USD x 10 <sup>6</sup> ] |
|---------------|---------------|--------------------------|------------|-----------------|-----------------|------------------------------------|---------------------------------|
| 1             | 2.0           | 304                      | 53.6%      | 9.4             | 6.1             | 5.6                                | -1.18                           |
| 5             | 12.3          | 1854                     | 59.5%      | 8.0             | 7.4             | 33.2                               | -0.07                           |
| 10            | 34.7          | 4670                     | 60.2%      | 7.2             | 9.2             | 83.7                               | 3.53                            |

Moving on to the configuration that employs pipelines as the energy storage medium, the results of the optimized configurations for the Chilean installation are reported in Figure 8.8 in terms of cumulative net present value (NPV). A marked difference with respect to the baseline case can be observed, with a significantly more favorable economic return on investment. The initial capital expenditure is reduced by approximately 30%, mainly due to the elimination of the dedicated storage tank. While for the 1 MW system size the end-of-life NPV reaches only about 0.1 million USD, with the payback occurring roughly one year before the end of the system lifetime, increasing the system scale leads to a substantially improved economic outlook. The break-even point shifts rapidly from

the 1 MW case to the 5 MW configuration, where it occurs at around 12 years, with a further improvement at 10 MW, for which it is reduced to approximately 10 years. The resulting NPVs for the 5 MW and 10 MW configurations are 6.7 and 17.22 million USD, respectively.

The main sizing and performance parameters of the optimized configurations are summarized in Table 8.3. It is worth noting that, for all three configurations, the discharging time reaches the maximum allowable value. This behavior is a direct consequence of the strongly stepwise electricity price trend in the Chilean market, which makes longer operating periods economically advantageous in order to maximize revenues. Moreover, when pipelines are used as the storage system, increasing the storage capacity leads to higher operating costs but does not affect capital investment, further justifying this design choice. The dimensions of the system components—namely compressor power and storage volume—increase more than linearly with output power, benefiting from economies of scale, particularly for the compressor. This scaling behavior enables shorter charging phases as the system size increases, thereby reducing charging costs and ultimately enhancing the overall return on investment.

In contrast, for the Danish installation, even under this configuration, an optimized configuration with positive cash flow cannot be achieved; for this reason, the results of the optimization are not reported.

Overall, these results indicate that the use of pipelines as energy storage systems, through the repurposing of existing infrastructure, represents an attractive option due to the reduction of a major cost component. However, this improvement alone is insufficient in contexts characterized by less pronounced electricity price variations, where the system still fails to ensure a satisfactory return on investment.

Finally, the results obtained by accounting for both the possibility of repurposing existing pipelines as compressed air storage systems and the concurrent presence of regulatory incentives are presented. This analysis was carried out exclusively for Denmark, since in the Chilean context the investment already yields strong economic performance; moreover, the electricity market price during peak hours is already high, making it



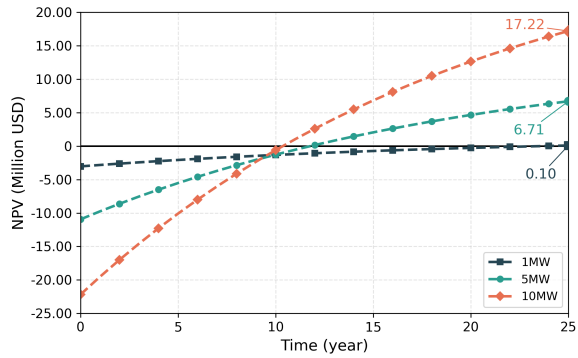


Figure 8.8: Cumulative NPV for the Chilean location with pipeline based storage

Table 8.3: Main sizing and performance parameters of the optimized pipeline-based storage configurations for the Chilean case study

| $P_t$<br>[MW] | $P_c$<br>[MW] | $V$<br>[m <sup>3</sup> ] | RTE<br>[%] | $t_{ch}$<br>[h] | $t_{ds}$<br>[h] | $E_{out}$<br>[ $\frac{MWh}{day}$ ] | NPV<br>[USD x 10 <sup>6</sup> ] |
|---------------|---------------|--------------------------|------------|-----------------|-----------------|------------------------------------|---------------------------------|
| 1             | 2.2           | 1202                     | 60.0%      | 9.6             | 10.0            | 7.4                                | 0.10                            |
| 5             | 12.1          | 5790                     | 62.1%      | 8.4             | 10.0            | 36.4                               | 6.71                            |
| 10            | 32.1          | 13206                    | 62.6%      | 6.9             | 10.0            | 78.5                               | 17.22                           |

unlikely that additional incentive schemes would be introduced or justified by the regulator.

For Denmark, a parametric analysis was conducted by varying the level of incentive, and only the results corresponding to incentive levels that ensure an investment payback within 20 years are reported. The outcomes are presented in terms of the net present value trend in Figure 8.9 and in terms of the main design and performance parameters of the selected configurations at the end of the optimization process in Table 8.4. The incentives required to meet this target amount to 180, 100, and 80 \$/MWh for system sizes of 1, 5, and 10 MW, respectively, representing substantial values; even in the most favorable case of the 10 MW system,

the incentive corresponds to an increase of approximately 65% over the market electricity selling price. Consequently, the regulatory intervention required to make this technology economically attractive is significant. For the 10 MW system operating at approximately 365 equivalent full cycles per year, the corresponding annual support translates into an annual remuneration on the order of 30 k\$/MWh of installed energy capacity. This level exceeds the remuneration observed in recent European capacity-based storage schemes on a per-capacity basis and reflects the challenging economic conditions faced by grid-connected CAES in electricity markets.

A trend consistent with previous findings is observed, with economic performance improving as system size increases, although the final NPV values remain limited, especially when compared to the initial investment. Furthermore, the results in Table 8.4 indicate that the optimal discharge duration is slightly above 4 hours, which coincides with the time window during which incentives are granted; this suggests that injecting energy into the market at prices lower than the incentivized level is not economically convenient.

Overall, this preliminary analysis highlights that the introduction of incentive schemes is essential to enhance the deployment of medium/small-scale CAES systems, as price arbitrage alone does not provide an adequate economic return. An additional opportunity may lie in participation in ancillary services markets, where CAES systems are well suited due to their inherent inertia and rapid start-up capabilities; however, this aspect is left for future developments of the present work.

*Table 8.4: Main sizing and performance parameters of the optimized configuration for the Danish location with pipeline-based storage under incentive conditions (break-even at 20 years)*

| $P_t$<br>[MW] | $P_c$<br>[MW] | $V$<br>[m <sup>3</sup> ] | RTE<br>[%] | $t_{ch}$<br>[h] | $t_{ds}$<br>[h] | $E_{out}$<br>[ $\frac{MWh}{day}$ ] | NPV<br>[USD × 10 <sup>6</sup> ] |
|---------------|---------------|--------------------------|------------|-----------------|-----------------|------------------------------------|---------------------------------|
| 1             | 1.3           | 511                      | 57.0%      | 7.9             | 4.8             | 3.3                                | 0.31                            |
| 5             | 6.5           | 2549                     | 62.4%      | 7.4             | 4.8             | 16.6                               | 0.82                            |
| 10            | 12.5          | 4519                     | 62.7%      | 6.8             | 4.3             | 29.4                               | 1.24                            |

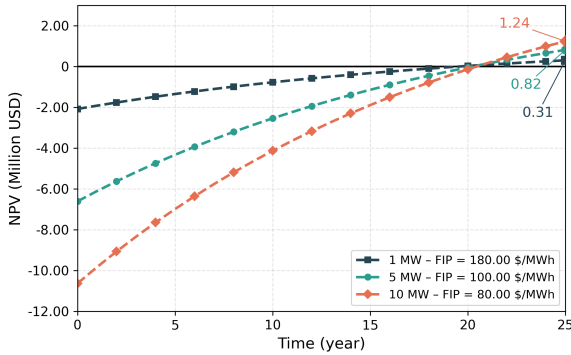


Figure 8.9: Cumulative NPV for the Danish location with pipeline based storage under incentives (break-even at 20 years)

## 8.4 Conclusions

This chapter assessed the techno-economic feasibility of grid-connected small-scale CAES systems operated under a simplified price-arbitrage strategy, with the objective of quantifying how market conditions and enabling measures influence system profitability. Starting from a baseline vessel-based configuration, two feasibility-enhancement pathways were investigated: (i) reducing the cost of the air storage subsystem through the repurposing of existing natural gas pipelines as compressed air reservoirs, and (ii) introducing a peak-oriented regulatory support mechanism in the form of a feed-in premium applied to the first four hours of discharge. The system was evaluated under two electricity market contexts characterized by different price dynamics. The Chilean market was selected as a limit case, representative of highly volatile electricity systems with pronounced off-peak to peak price differentials, providing favorable conditions for price-arbitrage operation. Conversely, the Danish market was analyzed as a more realistic deployment scenario for grid-connected storage, featuring significant intraday price fluctuations driven by high renewable penetration, albeit less extreme than those observed in the Chilean case.

The main findings of this chapter can be summarized as follows:

- The economic performance of grid-connected small-scale CAES systems is strongly influenced by electricity price dynamics. Highly volatile markets, such as the Chilean case, can enable arbitrage-based operation, while markets with significant but less extreme fluctuations, such as Denmark, remain more challenging without additional enabling mechanisms.
- System scale is a key driver of economic viability. For vessel-based storage, only the 10 MW configuration achieves a positive NPV (3.53 M\$) in the Chilean case, with a break-even time of approximately 22 years, whereas smaller systems (1–5 MW) remain non-profitable despite round-trip efficiencies exceeding 60%.
- The reuse of existing natural gas pipelines as compressed air storage substantially improves economic performance by reducing storage related capital costs, leading to an estimated ~30% CAPEX reduction and, in the Chilean case, enabling significantly higher NPVs, up to 17.22 M\$ for the 10 MW configuration, with break-even times reduced to 10 years.
- In electricity markets characterized by significant, yet non-extreme, price fluctuations, such as Denmark, economic feasibility cannot be achieved through price arbitrage alone and requires dedicated regulatory support mechanisms. In this context, the introduction of a peak-oriented feed-in premium enables payback within 20 years, but results in limited final profitability, with NPVs up to 1.24 M\$ for the 10 MW configuration, requiring incentive levels as high as 80–180 \$/MWh.

Overall, the results confirm that price arbitrage alone is generally insufficient to ensure attractive returns for small and medium-scale CAES systems. Economic viability is observed only in electricity markets characterized by extreme intraday price spreads, such as the Chilean case, or when cost reduction measures and regulatory incentives are jointly introduced, as in the Danish scenario.

These findings highlight the intrinsic limitations of a business model exclusively based on energy arbitrage for small-scale CAES and suggest that alternative operational paradigms should be explored to enhance system profitability. In this context, the introduction of dedicated incentive schemes appears to be a relevant enabling factor and is increasingly consistent with current market developments in several electricity systems, where regulatory frameworks are progressively evolving to explicitly remunerate flexibility and peak-oriented services. Moreover, CAES systems may be suitable for contributing to grid flexibility through the provision of ancillary services, which could complement revenues. While these aspects were not addressed within the scope of the present analysis, they emerge as relevant directions for future investigations aimed at improving the economic attractiveness of CAES storage solutions.

## Chapter 9

# Conclusions

This thesis investigated the role and realistic potential of small-scale Compressed Air Energy Storage systems, with a specific focus on their integration into decentralized energy systems. The work was motivated by the growing need for flexible, long-duration energy storage solutions capable of supporting renewable energy integration in both off-grid and grid-connected applications.

The main objective was to develop a coherent and comprehensive assessment framework combining time-varying thermodynamic modeling, exergy-based diagnostics, techno-economic optimization, and environmental evaluation. This integrated approach aimed to clarify the level of performance, efficiency, economic competitiveness, and environmental sustainability that can realistically be expected from small-scale CAES systems based on conventional and commercially available components.

The research activities were organized around three main progressive steps. First, a detailed modeling and diagnostic analysis of an existing micro-CAES prototype was performed to develop and validate a dynamic system-level model under realistic operating conditions, which subsequently served as a consistent reference framework for the analyses presented throughout this thesis. Second, sizing and optimization methodologies were developed for photovoltaic-integrated micro-CAES

systems, explicitly accounting for component interactions, off-design operation, and realistic load variability. Third, the analysis was extended to system-level applications, including hybrid PV-diesel-CAES systems for remote off-grid communities, the investigation of scaling effects and turbomachinery integration, and grid-connected CAES systems operating under market-driven conditions.

The main outcomes derived from this work are summarized in the following section.

## 9.1 Main outcomes

A dynamic, lumped-parameter thermodynamic model of an industrial micro compressed air energy storage prototype was developed using the Modelica language and validated against experimental data. The validated model provided a solid and reliable basis for all subsequent analyses, enabling a detailed energy and exergy assessment of system behavior under realistic operating conditions.

Building upon this validated modeling framework, an integrated sizing and optimization methodology was proposed, combining Morris screening for parameter reduction with an Efficient Global Optimization scheme based on adaptively updated Gaussian-process surrogates, fully coupled with the dynamic plant simulation. To the author's knowledge, this combination of surrogate-based Bayesian optimization with dynamic, off-design simulations has not been previously reported in the small-scale CAES context, where sizing studies typically rely on steady-state thermodynamic formulations, constant-efficiency component models, or full-enumeration parametric sweeps. This approach enabled an efficient exploration of the design space and the identification of cost-effective system configurations under techno-economic objectives. Despite relying on a detailed thermodynamic model, the methodology proved to be structured, robust, and computationally efficient, allowing for the systematic assessment of sizing parameters and the identification of optimal CAES designs. The analysis demonstrated that strong couplings between storage capacity, turboma-

chinery sizing, and maximum operating pressure govern both the economic performance and the operational feasibility of micro-CAES systems. The results confirmed that integrated thermo-economic optimization is essential to avoid oversizing penalties and misleading performance estimates. Within this framework, the sizing of the thermal energy recovery subsystem was found to have a comparatively limited impact on overall system performance. Consequently, a preliminary sizing strategy can be adopted for this subsystem, primarily aimed at preventing operational issues such as freezing, while limiting capital expenditure. The use of adaptively updated surrogate models allowed convergence toward optimal solutions within reasonable computational times, demonstrating the robustness and reliability of the proposed optimization framework. The optimized configurations exhibited round-trip efficiencies in the range of 30–40%, which, although relatively modest, are consistent with values typically reported for micro-CAES systems. The resulting levelized cost of storage exceeded 0.5 \$/kWh, a value higher than those reported for larger-scale CAES systems in the literature. This outcome is mainly attributable to the limited efficiency and small scale of the analyzed system, which inherently constrain economic performance.

The proposed methodology was further applied to explicitly investigate load variability and its effects on component performance, system efficiency, and optimal sizing. By comparing constant-load assumptions with realistic variable-load profiles, the analysis demonstrated that micro-CAES systems are intrinsically sensitive to off-design operation. The implemented exergy analysis revealed that the majority of exergy destruction and losses occur in the compressor and expanders, accounting for approximately 35% and 20% of the system input energy, respectively. Heat recovery and thermal storage systems play a secondary role, with a limited impact on overall efficiency. For the analyzed case study, operating a system optimized under constant-load assumptions in a highly variable load scenario led to an increase in expander exergy destruction of approximately 2%. High load variability also resulted in a measurable increase in the levelized cost of storage and a reduction in annual energy output, with round-trip



efficiency decreasing to approximately 28% when pursuing renewable self-sufficiency levels close to 100%. These results indicate that load variability particularly affects the expansion phase, forcing the expander to operate away from its nominal point and leading to increased irreversibilities.

Overall, these findings suggest that micro-CAES systems are best suited for applications characterized by stable or moderately variable loads, or for hybrid configurations in which load smoothing strategies can be implemented. Accurately accounting for load variability during the early design phase is therefore essential to avoid underestimating both performance losses and economic penalties. This further highlights the importance of the proposed sizing and optimization methodology for achieving reliable and realistic system designs.

The thesis further extended the analysis to off-grid applications by proposing and applying an integrated techno-economic and Life Cycle Assessment framework to hybrid PV-diesel-CAES systems for remote communities. A parametric life cycle inventory approach was developed and adopted, this approach represents an original contribution of the present work and introduced size-power correlations that enable the scaling of reference inventories, thereby better reflecting the behavior of commercial components and establishing a robust parametric foundation for future LCA-based assessments. The results demonstrated that integrating CAES into hybrid off-grid systems can significantly reduce greenhouse gas emissions compared to diesel-only solutions. Specifically, the carbon intensity of electricity generation decreased from nearly 1 kg CO<sub>2</sub>eq/kWh to values as low as 0.14 kg CO<sub>2</sub>eq/kWh under favorable operating conditions. The sequential integration of photovoltaic generation and CAES storage led to systematic improvements in environmental performance, despite an increase in impacts associated with raw material extraction and metal production. From an economic perspective, however, the integration of CAES at micro scale (approximately 10 kW) remains weakly competitive under current technological conditions. The levelized cost of electricity of the overall PV–diesel–CAES system typically ranges between 0.2 and 0.3 \$/kWh, depending on local diesel fuel prices. Only in a limited num-

ber of cases does this value fall below that of configurations in which photovoltaic generation is not coupled with energy storage. Economic feasibility was found to improve with increasing system size and in regions characterized by high diesel fuel costs, highlighting a strong dependence on local boundary conditions.

To address the efficiency limitations observed at small scale, the thesis investigated the effects of system scaling and the integration of radial turbomachinery. Map-based compressor and radial-turbine models were developed and enriched with a scale-dependent ceiling on the maximum achievable isentropic efficiency and a constraint on the maximum admissible rotational speed, capturing a performance reduction that, although structurally relevant for small-scale turbomachinery, has rarely been explicitly analyzed in the CAES literature. The optimal scheduling laws for inlet guide vanes and variable stator vanes were pre-computed offline and embedded in the plant model as non-dimensional look-up maps, preserving off-design accuracy while keeping system-level simulations tractable within the sizing optimization loop. The results showed that turbomachinery-based architectures enable substantial improvements in round-trip efficiency and cost indicators compared to configurations relying on positive-displacement machines.

Finally, the feasibility of grid-connected small-scale CAES systems operating under a simplified electricity price arbitrage strategy was assessed. The results indicated that economic performance is strongly influenced by both electricity price dynamics and system scale. Energy arbitrage alone was found to be insufficient to ensure profitability in moderately volatile markets, with positive economic outcomes observed only under highly volatile price conditions and at larger system scales. The reuse of existing infrastructure, such as natural gas pipelines repurposed for compressed air storage, was shown to significantly improve economic performance by reducing capital investment requirements. The analysis further demonstrated that, in markets characterized by appreciable but not extreme price fluctuations, dedicated regulatory support mechanisms are required to achieve economic feasibility. Within this context, the introduction of a

peak-oriented feed-in premium scheme was simulated. While this policy mechanism enables payback periods below 20 years, it requires relatively high incentive levels, ranging from approximately 80 to 180 \$/MWh, and results in limited overall profitability. Overall, these findings highlight the intrinsic limitations of business models exclusively based on energy arbitrage and suggest that alternative operational paradigms should be explored to enhance the economic attractiveness of CAES systems, such as the provision of grid flexibility and ancillary services.

Taken together, the results of this thesis provide a comprehensive and quantitatively grounded assessment of the performance, limitations, and application potential of micro and small-scale CAES systems. Overall, the findings indicate that small-scale CAES systems can deliver meaningful environmental benefits; however, their competitiveness remains constrained by intrinsic efficiency limitations, storage costs, and a strong sensitivity to operating conditions. A key outcome of this work is the demonstration that small-scale CAES should not be treated as a homogeneous technology class, as performance and feasibility strongly depend on system scale, component technology, and integration context.

The findings provide quantitative guidance for technology developers, system designers, and policymakers. Strategic priorities for expanding the application domain of CAES include: (i) development of high-efficiency and cost-effective small-scale turbomachinery, (ii) improved thermal management to reduce exergy destruction, (iii) hybrid architectures enabling load smoothing, and (iv) regulatory and market frameworks that appropriately value long-duration storage and grid services beyond energy arbitrage.

In conclusion, while small-scale CAES systems face inherent performance and cost challenges, this thesis shows that, under suitable technological and systemic conditions, they can represent a viable and environmentally beneficial component of future decentralized, low-carbon energy systems.

## 9.2 Limitations of the present work

Despite the comprehensive and integrated approach adopted in this thesis, some limitations should be acknowledged, mainly related to modeling assumptions, scope of application, and economic and environmental evaluation, in particular:

- The analysis relies on numerical models that require simplifying assumptions. Component performance is represented through characteristic maps derived from experimental data, manufacturer information, or literature correlations. These maps are assumed to be valid over the explored operating range and do not explicitly account for long-term effects (e.g., performance degradation), which may influence real-world operation over extended lifetimes.
- Economic and environmental assessments are based on literature cost correlations, representative market assumptions, and life cycle inventory data. As a result, outcomes depend on assumptions related to component costs, market conditions, component lifetimes, and end-of-life treatment. While these factors may influence absolute values, the comparative trends and relative performance of the analyzed configurations are considered robust.
- Representative load profiles, locations, and renewable resource data are assumed in the analyzed scenarios. While these assumptions are suitable for comparative and parametric analyses, they may not capture the full variability of real-world operating conditions across different geographical or socio-economic contexts.

## 9.3 Future research recommendations

Based on the findings and limitations of this thesis, the following directions are recommended for future research on small-scale CAES

systems:

- **Advanced system configurations and control strategies.** Future research should investigate advanced control strategies, multi-pressure storage concepts, and hybrid configurations (e.g. CAES coupled with small electrochemical storage) to mitigate the impact of load variability. The integration of CAES within hybrid storage systems and advanced energy management frameworks represents a promising pathway to improve operational flexibility and economic performance.
- **Refinement of techno-economic and environmental models.** From an economic perspective, future work should improve the accuracy of cost functions through calibration against updated market data and real-world case studies. From an environmental standpoint, more detailed life cycle inventories of system components should be developed, moving beyond the assumptions adopted in this thesis.
- **Long-term performance and market evolution.** The integration of component degradation models and performance decay curves would enable a more realistic assessment of long-term system behavior and economic viability. In addition, the explicit consideration of evolving electricity market structures would provide deeper insight into the future role of small-scale CAES systems.
- **Additional revenue streams for grid-connected applications.** Future studies should explore alternative revenue mechanisms beyond energy arbitrage, including ancillary service provision, capacity markets, and hybrid market participation, in order to assess the conditions under which grid-connected CAES systems may become economically competitive.
- **Life cycle assessment of energy storage technologies.** The life cycle assessment framework could be extended to include a comprehensive comparison with alternative energy storage technologies,

considering both economic and environmental dimensions. In particular, comparisons with other thermomechanical storage options, such as pumped thermal energy storage systems, as well as with electrochemical solutions. Such an analysis would allow a more robust positioning of small-scale CAES with respect to competing storage solutions.

- **Generalization and cross-context application** Future work may extend the application of the proposed modeling, sizing, and optimization framework to a wider range of geographical and regulatory contexts. While the present study focuses on selected representative off-grid and grid-connected scenarios, the methodology can be applied to different electricity markets, climatic conditions, and energy mixes. Such extensions could support the development of feasibility maps highlighting regions or application domains where small-scale CAES systems are more likely to be technically and economically viable.



## Appendix A – Life Cycle Inventory

As described in the main text, the environmental assessment relies on scaling the Life Cycle Inventories (LCI) according to a characteristic parameter representative of each component. The inventories reported in this appendix are expressed per unit of reference: 1 kg for the compressor, expander, electric motor, thermal storage, lubricating oil, and heat exchanger; 1 kW of installed peak power for the photovoltaic system; and 1 kg of diesel consumed during the operation of the diesel generators. Negative flows represent end-of-life credits modelled according to the cut-off approach. Data originate from scientific literature or Ecoinvent datasets, with full references provided in the main text.

| Component               | Type   | Flow                                      | Amount      | Unit           |
|-------------------------|--------|-------------------------------------------|-------------|----------------|
| <b>Air vessel</b>       | Input  | Iron scrap, unsorted                      | -0.85       | kg             |
|                         |        | Steel, low alloyed, hot rolled            | 1           | kg             |
|                         |        | Welding, arc, steel                       | 0.006878505 | m              |
|                         |        | Water                                     | 0.008691589 | m <sup>3</sup> |
|                         | Output | Air vessel                                | 1           | kg             |
|                         |        | Scrap steel                               | 0.15        | kg             |
| <b>Diesel generator</b> | Input  | aluminium scrap, post-consumer            | -0.2625     | kg             |
|                         |        | aluminium, cast alloy                     | 0.169642857 | kg             |
|                         |        | aluminium, wrought alloy                  | 0.180357143 | kg             |
|                         |        | copper scrap, sorted, pressed             | -0.011      | kg             |
|                         |        | copper, cathode                           | 0.019642857 | kg             |
|                         |        | electricity, medium voltage               | 0.016071429 | MJ             |
|                         |        | heat, district or industrial, natural gas | 0.053571429 | MJ             |
|                         |        | iron scrap, unsorted                      | -0.51       | kg             |
|                         |        | polystyrene, high impact                  | 0.030357143 | kg             |
|                         |        | steel, low-alloyed                        | 0.6         | kg             |



| Component                             | Type   | Flow                                 | Amount      | Unit |
|---------------------------------------|--------|--------------------------------------|-------------|------|
|                                       | Output | Diesel generator                     | 1           | kg   |
|                                       |        | scrap copper                         | 0.008642857 | kg   |
|                                       |        | scrap steel                          | 0.09        | kg   |
|                                       |        | waste aluminium                      | 0.0875      | kg   |
|                                       |        | waste polystyrene                    | 0.030357143 | kg   |
| <b>Diesel use<br/>(kW &lt; 18.64)</b> | Input  | Building machine                     | 5.73E-06    | item |
|                                       |        | Diesel                               | 1           | kg   |
|                                       |        | Lubricating oil                      | 0.003828    | kg   |
|                                       |        | Lubricating oil                      | 0.018172    | kg   |
|                                       | Output | Oxygen                               | 2.320475    | kg   |
|                                       |        | 2,2,4-Trimethylpentane               | 1.83E-06    | kg   |
|                                       |        | Acenaphthene                         | 1.51E-07    | kg   |
|                                       |        | Acenaphthylene                       | 1.27E-07    | kg   |
|                                       |        | Acetaldehyde                         | 0.000135    | kg   |
|                                       |        | Acrolein                             | 7.72E-06    | kg   |
|                                       |        | Ammonia                              | 2.60E-05    | kg   |
|                                       |        | Anthracene                           | 6.48E-10    | kg   |
|                                       |        | Benz(a)anthracene                    | 1.07E-09    | kg   |
|                                       |        | Benzene                              | 5.18E-05    | kg   |
|                                       |        | Benzo(a)pyrene                       | 5.28E-10    | kg   |
|                                       |        | Benzo(b)fluoranthene                 | 7.39E-10    | kg   |
|                                       |        | Benzo(ghi)perylene                   | 2.86E-10    | kg   |
|                                       |        | Benzo(k)fluoranthene                 | 5.28E-10    | kg   |
|                                       |        | Butadiene                            | 4.74E-06    | kg   |
|                                       |        | Cadmium II                           | 1.00E-08    | kg   |
|                                       |        | Carbon dioxide, fossil               | 3.158805    | kg   |
|                                       |        | Carbon monoxide, fossil              | 0.011764    | kg   |
|                                       |        | Chromium III                         | 1.18E-10    | kg   |
|                                       |        | Chromium VI                          | 6.07E-11    | kg   |
|                                       |        | Chrysene                             | 2.86E-09    | kg   |
|                                       |        | Copper ion                           | 1.70E-06    | kg   |
|                                       |        | Dibenz(a,h)anthracene                | 4.37E-12    | kg   |
|                                       |        | Dinitrogen monoxide                  | 0.00012     | kg   |
|                                       |        | Dioxins (as 2,3,7,8-TCDD)            | 6.00E-13    | kg   |
|                                       |        | Ethane                               | 2.42E-06    | kg   |
|                                       |        | Ethylbenzene                         | 7.90E-06    | kg   |
|                                       |        | Fluoranthene                         | 2.56E-08    | kg   |
|                                       |        | Fluorene                             | 1.51E-07    | kg   |
|                                       |        | Formaldehyde                         | 0.000301    | kg   |
|                                       |        | Hexane                               | 4.05E-06    | kg   |
|                                       |        | Indeno(1,2,3-cd)pyrene               | 1.19E-10    | kg   |
|                                       |        | Machine operation, diesel < 18.64 kW | 0.907178    | h    |
|                                       |        | Manganese II                         | 8.16E-09    | kg   |
|                                       |        | Methane, fossil                      | 3.87E-05    | kg   |
|                                       |        | Naphthalene                          | 6.94E-07    | kg   |
|                                       |        | Nickel II                            | 1.21E-08    | kg   |
|                                       |        | Nitrogen oxides                      | 0.024123    | kg   |
|                                       |        | NM VOC                               | 0.001936    | kg   |
|                                       |        | PM2.5                                | 0.001463    | kg   |

| Component                                  | Type   | Flow                                                     | Amount   | Unit |
|--------------------------------------------|--------|----------------------------------------------------------|----------|------|
|                                            |        | PM10                                                     | 9.61E-05 | kg   |
|                                            |        | PM2.5–10                                                 | 4.52E-05 | kg   |
|                                            |        | Phenanthrene                                             | 3.92E-07 | kg   |
|                                            |        | Propanal                                                 | 3.01E-05 | kg   |
|                                            |        | Pyrene                                                   | 4.37E-09 | kg   |
|                                            |        | Selenium IV                                              | 1.00E-08 | kg   |
|                                            |        | Styrene                                                  | 1.51E-06 | kg   |
|                                            |        | Sulfur dioxide                                           | 2.93E-05 | kg   |
|                                            |        | Toluene                                                  | 3.81E-05 | kg   |
|                                            |        | Waste mineral oil                                        | 0.022    | kg   |
|                                            |        | Xylenes                                                  | 2.69E-05 | kg   |
|                                            |        | Zinc II                                                  | 1.00E-06 | kg   |
| <b>Diesel use<br/>(18.64–74.57<br/>kW)</b> | Input  | diesel                                                   | 1        | Kg   |
|                                            |        | lubricating oil                                          | 0.021996 | kg   |
|                                            | Output | Oxygen                                                   | 2.319766 | kg   |
|                                            |        | 2,2,4-Trimethylpentane                                   | 1.26E-06 | kg   |
|                                            |        | Acenaphthene                                             | 1.28E-07 | kg   |
|                                            |        | Acenaphthylene                                           | 1.08E-07 | kg   |
|                                            |        | Acetaldehyde                                             | 9.32E-05 | kg   |
|                                            |        | Acrolein                                                 | 5.32E-06 | kg   |
|                                            |        | Ammonia                                                  | 2.6E-05  | kg   |
|                                            |        | Anthracene                                               | 5.51E-10 | kg   |
|                                            |        | Benz(a)anthracene                                        | 9.09E-10 | kg   |
|                                            |        | Benzene                                                  | 3.57E-05 | kg   |
|                                            |        | Benzo(a)pyrene                                           | 4.48E-10 | kg   |
|                                            |        | Benzo(b)fluoranthene                                     | 6.27E-10 | kg   |
|                                            |        | Benzo(ghi)perylene                                       | 2.43E-10 | kg   |
|                                            |        | Benzo(k)fluoranthene                                     | 4.48E-10 | kg   |
|                                            |        | Butadiene                                                | 3.27E-06 | kg   |
|                                            |        | Cadmium II                                               | 1E-08    | kg   |
|                                            |        | Carbon dioxide, fossil                                   | 3.162173 | kg   |
|                                            |        | Carbon monoxide, fossil                                  | 0.008996 | kg   |
|                                            |        | Chromium III                                             | 1.18E-10 | kg   |
|                                            |        | Chromium VI                                              | 6.07E-11 | kg   |
|                                            |        | Chrysene                                                 | 2.43E-09 | kg   |
|                                            |        | Copper ion                                               | 1.7E-06  | kg   |
|                                            |        | Dibenz(a,h)anthracene                                    | 3.71E-12 | kg   |
|                                            |        | Dinitrogen monoxide                                      | 0.00012  | kg   |
|                                            |        | Dioxins, measured as 2,3,7,8 tetrachlorodibenzo-p-dioxin | 6E-13    | kg   |
|                                            |        | Ethane                                                   | 1.67E-06 | kg   |
|                                            |        | Ethyl benzene                                            | 5.44E-06 | kg   |
|                                            |        | Fluoranthene                                             | 2.18E-08 | kg   |
|                                            |        | Fluorene                                                 | 1.28E-07 | kg   |
|                                            |        | Formaldehyde                                             | 0.000207 | kg   |
|                                            |        | Hexane                                                   | 2.79E-06 | kg   |
|                                            |        | Indeno(1,2,3-cd)pyrene                                   | 1.01E-10 | kg   |

| Component                                  | Type   | Flow                                                                     | Amount   | Unit |
|--------------------------------------------|--------|--------------------------------------------------------------------------|----------|------|
|                                            |        | machine operation, diesel, $\geq 18.64$ kW                               | 0.216497 | h    |
|                                            |        | and $< 74.57$ kW, generators                                             |          |      |
|                                            |        | Manganese II                                                             | 8.16E-09 | kg   |
|                                            |        | Methane, fossil                                                          | 2.67E-05 | kg   |
|                                            |        | Naphthalene                                                              | 5.89E-07 | kg   |
|                                            |        | Nickel II                                                                | 1.21E-08 | kg   |
|                                            |        | Nitrogen oxides                                                          | 0.022099 | kg   |
|                                            |        | NM VOC, non-methane volatile organic compounds                           | 0.001335 | kg   |
|                                            |        | Particulate Matter, $\leq 2.5$ $\mu\text{m}$                             | 0.001242 | kg   |
|                                            |        | Particulate Matter, $\leq 10$ $\mu\text{m}$                              | 8.17E-05 | kg   |
|                                            |        | Particulate Matter, $\leq 2.5$ $\mu\text{m}$ and $\leq 10$ $\mu\text{m}$ | 3.84E-05 | kg   |
|                                            |        | Phenanthrene                                                             | 3.33E-07 | kg   |
|                                            |        | Propanal                                                                 | 2.07E-05 | kg   |
|                                            |        | Pyrene                                                                   | 3.71E-09 | kg   |
|                                            |        | Selenium IV                                                              | 1E-08    | kg   |
|                                            |        | Styrene                                                                  | 1.04E-06 | kg   |
|                                            |        | Sulfur dioxide                                                           | 2.93E-05 | kg   |
|                                            |        | Toluene                                                                  | 2.63E-05 | kg   |
|                                            |        | waste mineral oil                                                        | 0.022002 | kg   |
|                                            |        | Xylenes, unspecified                                                     | 1.86E-05 | kg   |
|                                            |        | Zinc II                                                                  | 1E-06    | kg   |
| <b>Diesel use<br/>(18.64–74.57<br/>kW)</b> | Input  | diesel                                                                   | 1        | Kg   |
|                                            |        | lubricating oil                                                          | 0.021996 | kg   |
|                                            | Output | Oxygen                                                                   | 2.319766 | kg   |
|                                            |        | 2,2,4-Trimethylpentane                                                   | 1.26E-06 | kg   |
|                                            |        | Acenaphthene                                                             | 1.28E-07 | kg   |
|                                            |        | Acenaphthylene                                                           | 1.08E-07 | kg   |
|                                            |        | Acetaldehyde                                                             | 9.32E-05 | kg   |
|                                            |        | Acrolein                                                                 | 5.32E-06 | kg   |
|                                            |        | Ammonia                                                                  | 2.6E-05  | kg   |
|                                            |        | Anthracene                                                               | 5.51E-10 | kg   |
|                                            |        | Benz(a)anthracene                                                        | 9.09E-10 | kg   |
|                                            |        | Benzene                                                                  | 3.57E-05 | kg   |
|                                            |        | Benzo(a)pyrene                                                           | 4.48E-10 | kg   |
|                                            |        | Benzo(b)fluoranthene                                                     | 6.27E-10 | kg   |
|                                            |        | Benzo(ghi)perylene                                                       | 2.43E-10 | kg   |
|                                            |        | Benzo(k)fluoranthene                                                     | 4.48E-10 | kg   |
|                                            |        | Butadiene                                                                | 3.27E-06 | kg   |
|                                            |        | Cadmium II                                                               | 1E-08    | kg   |
|                                            |        | Carbon dioxide, fossil                                                   | 3.162173 | kg   |
|                                            |        | Carbon monoxide, fossil                                                  | 0.008996 | kg   |
|                                            |        | Chromium III                                                             | 1.18E-10 | kg   |
|                                            |        | Chromium VI                                                              | 6.07E-11 | kg   |
|                                            |        | Chrysene                                                                 | 2.43E-09 | kg   |
|                                            |        | Copper ion                                                               | 1.7E-06  | kg   |
|                                            |        | Dibenz(a,h)anthracene                                                    | 3.71E-12 | kg   |
|                                            |        | Dinitrogen monoxide                                                      | 0.00012  | kg   |

| Component      | Type   | Flow                                                              | Amount    | Unit |
|----------------|--------|-------------------------------------------------------------------|-----------|------|
|                |        | Dioxins, measured as 2,3,7,8-tetrachlorodibenzo-p-dioxin          | 6E-13     | kg   |
|                |        | Ethane                                                            | 1.67E-06  | kg   |
|                |        | Ethyl benzene                                                     | 5.44E-06  | kg   |
|                |        | Fluoranthene                                                      | 2.18E-08  | kg   |
|                |        | Fluorene                                                          | 1.28E-07  | kg   |
|                |        | Formaldehyde                                                      | 0.000207  | kg   |
|                |        | Hexane                                                            | 2.79E-06  | kg   |
|                |        | Indeno(1,2,3-cd)pyrene                                            | 1.01E-10  | kg   |
|                |        | machine operation, diesel, >= 18.64 kW and < 74.57 kW, generators | 0.216497  | h    |
|                |        | Manganese II                                                      | 8.16E-09  | kg   |
|                |        | Methane, fossil                                                   | 2.67E-05  | kg   |
|                |        | Naphthalene                                                       | 5.89E-07  | kg   |
|                |        | Nickel II                                                         | 1.21E-08  | kg   |
|                |        | Nitrogen oxides                                                   | 0.022099  | kg   |
|                |        | NM VOC, non-methane volatile organic compounds                    | 0.001335  | kg   |
|                |        | Particulate Matter, > 2.5 um                                      | 0.001242  | kg   |
|                |        | Particulate Matter, < 10 um                                       | 8.17E-05  | kg   |
|                |        | Particulate Matter, < 2.5 um and > 10um                           | 3.84E-05  | kg   |
|                |        | Phenanthrene                                                      | 3.33E-07  | kg   |
|                |        | Propanal                                                          | 2.07E-05  | kg   |
|                |        | Pyrene                                                            | 3.71E-09  | kg   |
|                |        | Selenium IV                                                       | 1E-08     | kg   |
|                |        | Styrene                                                           | 1.04E-06  | kg   |
|                |        | Sulfur dioxide                                                    | 2.93E-05  | kg   |
|                |        | Toluene                                                           | 2.63E-05  | kg   |
|                |        | waste mineral oil                                                 | 0.022002  | kg   |
|                |        | Xylenes, unspecified                                              | 1.86E-05  | kg   |
|                |        | Zinc II                                                           | 1E-06     | kg   |
| Electric motor | Input  | Aluminium, cast alloy                                             | 0.114531  | kg   |
|                |        | Alluminium scrap, post-consume                                    | -0.085898 | kg   |
|                |        | Copper scrap, sorted, pressed                                     | -0.11040  | kg   |
|                |        | Copper, cathode                                                   | 0.197117  | kg   |
|                |        | Steel, low alloyed                                                | 0.687963  | kg   |
|                |        | Electricity, medium voltage                                       | 2.635762  | MJ   |
|                |        | Iron scrap, unsorted                                              | 0.584924  | kg   |
|                |        | Sheet rolling, steel                                              | 0.687963  | kg   |
|                | Output | Wire drawing, copper                                              | 0.197117  | kg   |
|                |        | Electric motor                                                    | 1         | kg   |
|                |        | Scrap copper                                                      | 0.086717  | kg   |
|                |        | Scrap steel                                                       | 0.274032  | kg   |
|                |        | Waste aluminium                                                   | 0.032917  | kg   |
| Heat exchanger | Input  | Chromium steel pipe                                               | 0.2       | kg   |
|                |        | Iron scrap, unsorted                                              | -0.84995  | kg   |
|                |        | Iron-nickel-chromium alloy                                        | 0.111111  | kg   |
|                |        | Steel, low alloyed                                                | 0.777778  | kg   |

| Component                | Type   | Flow                                                   | Amount      | Unit    |
|--------------------------|--------|--------------------------------------------------------|-------------|---------|
|                          | Output | Metal working, average for metal product manufacturing | 0.111111    | kg      |
|                          |        | Heat exchanger                                         | 1           | kg      |
|                          |        | Scrap steel                                            | 0.928729    | kg      |
| Mineral oil              | Input  | Lubricating oil                                        | 1           | Kg      |
|                          | Output | Lubricating oil                                        | 1           | kg      |
|                          |        | Waste lubricating oil                                  | 1           | Kg      |
| Photovoltaic system      | Input  | Diesel, burned in building machine                     | 13.4614     | MJ      |
|                          |        | Electricity, low voltage                               | 0.063211    | kWh     |
|                          |        | Inverter, 500 kW                                       | 0.005491    | item    |
|                          |        | Photovoltaic mounting system (570 kWp)                 | 7.497368    | m2      |
|                          | Output | Photovoltaic panel, multi-Si wafer                     | 7.722281    | m2      |
|                          |        | PV electric installation (570 kWp)                     | 0.001754    | item    |
|                          |        | Photovoltaic system                                    | 1           | kWp     |
|                          |        |                                                        |             |         |
| Reciprocating compressor | Input  | aluminium scrap, new                                   | -           | kg      |
|                          |        |                                                        | 0.123687273 |         |
|                          |        | aluminium, cast alloy                                  | 0.618436364 | kg      |
|                          |        | copper, cathode                                        | 0.007272727 | kg      |
|                          |        | electricity, medium voltage                            | 2.028122585 | kWh     |
|                          |        | ethylene glycol                                        | 0.02400311  | kg      |
|                          |        | ethylene glycol                                        | 0.001451435 | kg      |
|                          |        | heat, district or industrial, natural gas              | 2.103940252 | MJ      |
|                          |        | heat, district or industrial, other than natural gas   | 0.059706413 | MJ      |
|                          |        | injection moulding                                     | 0.048       | kg      |
|                          |        | iron scrap, unsorted                                   | -           | kg      |
|                          |        |                                                        | 0.078690909 |         |
|                          |        | lead                                                   | 0.047272727 | kg      |
|                          |        | lubricating oil                                        | 0.018021818 | kg      |
|                          |        | lubricating oil                                        | 0.003796364 | kg      |
|                          |        | palladium                                              | 1.09E-06    | kg      |
|                          |        | platinum                                               | 5.82E-06    | kg      |
|                          |        | polyethylene, high density, granulate                  | 0.056       | kg      |
|                          |        | polyphenylene sulfide                                  | 0.1         | kg      |
|                          |        | reinforcing steel                                      | 0.152727273 | kg      |
|                          |        | road vehicle factory                                   | 2.76E-10    | Item(s) |
|                          |        | steel, low-alloyed, hot rolled                         | 0.393436364 | kg      |
|                          |        | sulfuric acid                                          | 0.003215754 | kg      |
|                          |        | sulfuric acid                                          | 0.000420609 | kg      |
|                          |        | synthetic rubber                                       | 0.010909091 | kg      |
|                          |        | tap water                                              | 3.051661087 | kg      |
|                          |        | wire drawing, copper                                   | 0.007272727 | kg      |
|                          | Output | Reciprocating compressor                               | 1           | kg      |
|                          |        | NMVOC, non-methane volatile organic compounds          | 0.00454906  | kg      |
|                          |        | Phosphate                                              | 9.48E-07    | kg      |
|                          |        | TOC, Total Organic Carbon                              | 6.77E-05    | kg      |

| Component                  | Type    | Flow                                           | Amount      | Unit |
|----------------------------|---------|------------------------------------------------|-------------|------|
|                            |         | used internal combustion engine, passenger car | 1           | kg   |
|                            |         | waste plastic, mixture                         | 0.014181818 | kg   |
|                            |         | Water                                          | 0.000457749 | m3   |
|                            |         | Water, GLO                                     | 0.002593912 | m3   |
|                            |         | BOD5, Biological Oxygen Demand                 | 2.46E-05    | kg   |
|                            |         | COD, Chemical Oxygen Demand                    | 0.00018291  | kg   |
|                            |         | DOC, Dissolved Organic Carbon                  | 6.77E-05    | kg   |
| <b>Thermal storage</b>     | Inputs  | Alkyd paint, white                             | 0.003559    | kg   |
|                            |         | Electricity, medium voltage                    | 0.441441    | MJ   |
|                            |         | Glass wool mat                                 | 0.070678    | kg   |
|                            |         | Iron scrap, unsorted                           | -0.78099    | kg   |
|                            |         | Lubricating oil                                | 2.254237    | kg   |
|                            |         | Natural gas                                    | 0.540678    | m3   |
|                            |         | Polyvinylchloride                              | 0.007034    | kg   |
|                            |         | Steel, chromium steel 18/8                     | 0.141356    | kg   |
|                            | Outputs | Steel, low alloyed                             | 0.777458    | kg   |
|                            |         | Welding, arc, steel                            | 0.027288    | m    |
|                            |         | Thermal storage                                | 1           | kg   |
|                            |         | Scrap steel                                    | 0.137797    | kg   |
|                            |         | Waste mineral oil                              | 2.254237    | kg   |
|                            |         | Waste mineral wool                             | 0.070678    | kg   |
|                            |         | Waste paint on metal                           | 0.003559    | kg   |
|                            |         | Waste polyvinylchloride                        | 0.007034    | kg   |
| <b>Volumetric expander</b> | Input   | aluminium scrap, post-consumer                 | -0.1071428  | kg   |
|                            |         | aluminium, cast alloy                          | 0.0457142   | kg   |
|                            |         | aluminium, wrought alloy                       | 0.09714285  | kg   |
|                            |         | cast iron                                      | 0.21428571  | kg   |
|                            |         | copper scrap, sorted, pressed                  | -0.08       | kg   |
|                            |         | copper, cathode                                | 0.1428571   | kg   |
|                            |         | injection moulding                             | 0.0714285   | kg   |
|                            |         | iron scrap, unsorted                           | -0.522142   | kg   |
|                            |         | polystyrene, high impact                       | 0.07142851  | kg   |
|                            |         | printed wiring board, Pb containing            | 0.000896429 | kg   |
|                            |         | printed wiring board, Pb free                  | 0.000896429 | kg   |
|                            |         | sheet rolling, aluminium                       | 0.142857143 | kg   |
|                            |         | sheet rolling, chromium steel                  | 0.114285714 | kg   |
|                            |         | sheet rolling, steel                           | 0.285714286 | kg   |
|                            |         | steel, chromium steel 18/8, hot rolled         | 0.114285714 | kg   |
|                            |         | steel, low-alloyed, hot rolled                 | 0.285714286 | kg   |
|                            |         | synthetic rubber                               | 0.021428571 | kg   |
|                            |         | wire drawing, copper                           | 0.142857143 | kg   |
|                            | Output  | Volumetric expander                            | 1           | kg   |
|                            |         | scrap copper                                   | 0.062857143 | kg   |
|                            |         | scrap steel                                    | 0.092142857 | kg   |
|                            |         | used printed wiring boards                     | 0.001792857 | kg   |
|                            |         | waste aluminium                                | 0.035714286 | kg   |
|                            |         | waste polystyrene                              | 0.071428571 | kg   |

waste rubber

0.021428571 kg

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La borsa di dottorato cofinanziata con risorse dell'Unione europea-*NextGeneration EU*  
Piano Nazionale di Ripresa e Resilienza Missione 4 – Componente 2 “Dalla Ricerca all'Impresa” –  
Investimento 3.3 “Introduzione di dottorati innovativi che rispondono ai fabbisogni di innovazione delle  
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